NUCLEAR VS. FOSSIL PROPULSION FOR ICEBREAKERS

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FOR ICEBREAKERS

by

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ABSTRACT

The economic desirability of utilizing nuclear, in lieu of conventional, propulsion for polar icebreakers is investigated. Mission requirements and operating theater are defined. An investigation of current empirical, semi-empirical, and theoretical ice resistance prediction techniques for ships is conducted. The theoretical approach is selected to evaluate the icebreaking capabilities of ship systems developed in succeeding sections.

A suitable fossil-fueled propulsion plant is identified. A simple weight balance math synthesis model is formulated, and a ship system solution determined. The icebreaking capabilities are determined and checked against the mission requirements to establish solution validity.

A nuclear-fueled propulsion plant is selected. The math model is modified to reflect the nuclear impact. A nuclear solution is identified and validated.

An economic comparison of the nuclear and fossil ships is performed. Capital costs and annual operating costs are derived for each ship. Discounted cash flow techniques are employed to produce an Annual Equivalent Cash Flow (AECF) for each ship. The lowest AECF is indicative of the economically superior propulsion plant. Finally, discount rates and fuel prices are varied to investigate solution sensitivity.

Thesis supervisors:	Professor Philip Mandel Professor Henry S. Marcus
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CHAPTER I

INTRODUCTION

1.1 Introduction

The objective of this thesis is to compare nuclear versus fossil fuel propulsion for polar icebreakers. The mission requirements set forth by the United States Coast Guard in 1966 will serve as the basis for the ship system developments. These requirements are contained in "Operational Concepts and Requirements for Replacement of Polar Icebreakers" [1]. The thesis will evaluate the propulsion plant powering question in light of recent radical changes in fuel prices, and ten years of technology and experience.

The goal will be attained by: establishing the mission requirements and operating theater (Chapter I), investigating and selecting suitable ice resistance calculation techniques (Chapter II), selecting a fossil-fueled propulsion plant and formulating the resultant ship system (Chapter III), selecting a nuclear-fueled propulsion plant and formulating the resultant ship system (Chapter IV), and, finally, performing an economic comparison of the nuclear and fossil ship systems (Chapter V). The goal of the analysis is the identification of an optimum economic choice of ship systems.

1.2 Mission requirements

In 1966, the U.S.C.G. investigated possible production of six icebreakers. Four of these vessels would be capable of breaking 6-7 feet of ice in the continuous mode. Continuous mode is generally accepted to be no less than 3-4 knots. The mission for these vessels was described as a combination ice escort/logistic resupply/scientific research. The remaining two vessels were to be capable of breaking 9-10 feet of ice in the continuous mode. These vessels were to be "especially designed to conduct military operations and scientific investigations deep within heretofore inaccessible polar areas" [1]. The endurance requirements for both types of icebreakers consisted of a 77 day profile, equivalent to 48 full power days without refueling. An open water speed of 17 knots was required for both ships. However, no specific ramming requirement was established.

In 1967, the U.S.C.G. elected to build a two vessel class of ships, the POLAR Class. The endurance requirement was reduced to 28.5 full power days. The ice escort/logistic resupply/scientific research vessel icebreaking capability was preserved (i.e. 6 feet of ice in the continuous mode). The open water speed requirement of 17 knots was maintained.

Since 1967, the U.S. Coast Guard's role in the Arctic has increased dramatically. One manifestation of this

phenomenon is the increased need for icebreaking services. The changing world political/economical situation has created a high premium on natural resources. Therefore, the urgency to develop the Arctic has necessitated increased utilization of the icy Arctic waterways. Consequently, the author feels that the original endurance of 48 full power days should be reconsidered. The six foot continuous mode icebreaking and open water speed (17 knots) requirements will be held constant.

1.3 Operating theater

German [2] states that one of the primary considerations in icebreaker design is the "operating theater and the season". In view of the increased demand for icebreaking services from the commercial sector of the economy, the trade routes to Prudhoe Bay will be considered a probable operating environment.

Clearly, as a member of the U.S. Coast Guard's icebreaking fleet, the vessel must be capable of Antarctic operations. The importance of endurance is obvious in this connotation. However, Antarctic-related missions will not be considered due to lack of data and commercial impetus (Anarctica is an international scientific zone).

The Canadian government has enacted the Arctic Waters Pollution Prevention Act, the Canadian Government Arctic Shipping Pollution Regulations, and the "Shipping Safety

Control Zones Order," providing guidance in the development of Arctic transportation. The Canadian Arctic has been divided into sixteen zones (Figure 1.1) [4], and a "classification of vessels to conform with specified operating seasons in each of the zones" was established. The regulations pertain to vessel construction and powering, assigning vessels to Arctic Classes from 1 to 10. The Class number is an indication of the expected thickness of ice that the vessel can break in the continuous mode (number and thickness being approximately equal). A convenient means of representing the seasonal and zonal limitation of ship classes is presented in Table 1.1 [5]. The Northwest Passage and Bering Strait trade routes [4] are plotted on zonal charts to indicate the appropriate zones (Figures 1.2 and 1.3 respectively). The zones in Figure 1.3 (Bering Strait Route) are an extension of the Canadian system, performed by the Arctic Institute of North America [4].

The seasonal operating constraints implied by the Canadian Classification system are summarized in Table 1.2. The table contains the seasonal limitations of a Class 3 (Wind Class), Class 6 (Polar Class), and Class 9 (the military/scientific ship the U.S.C.G. elected not to build), for each of the trade routes. This table is constructed from data contained in Table 1.1. It demonstrates the improvement in seasonal capabilities for the

Figure 1.1
Canadian Ice Control Zones

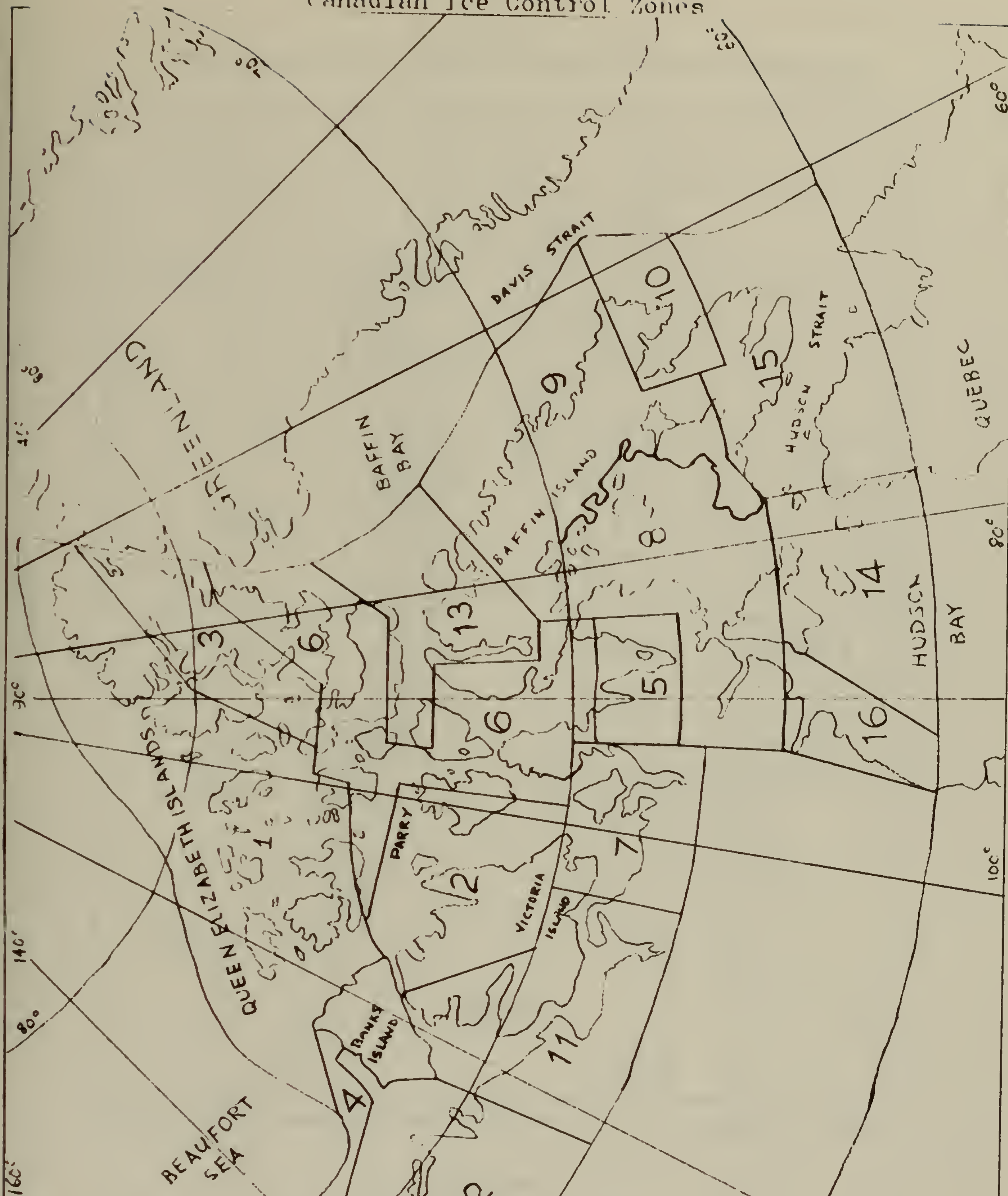


Table 1.1

Canadian Arctic Zonal Classification Nomogram

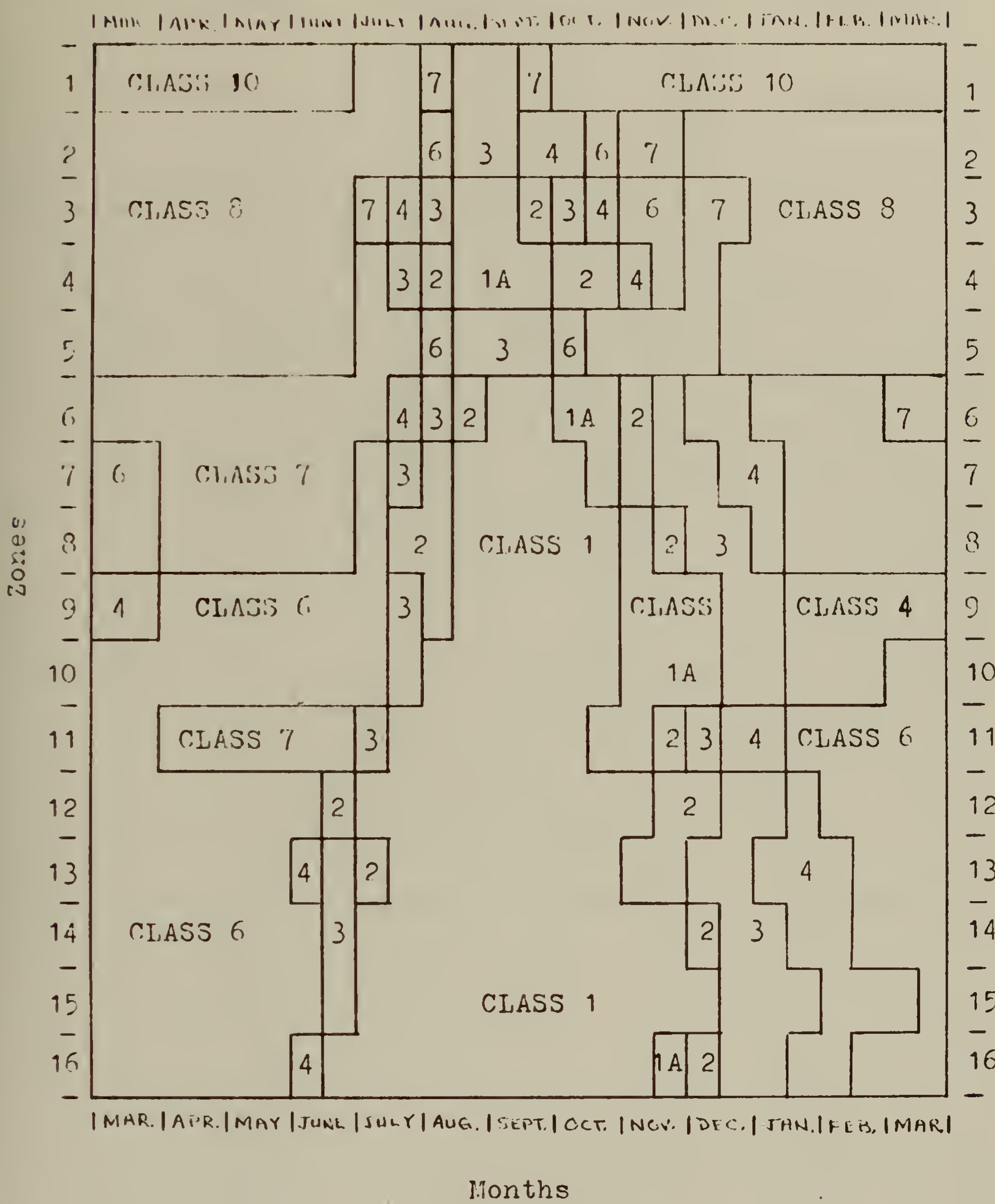


Figure 1.2

Northwest Passage

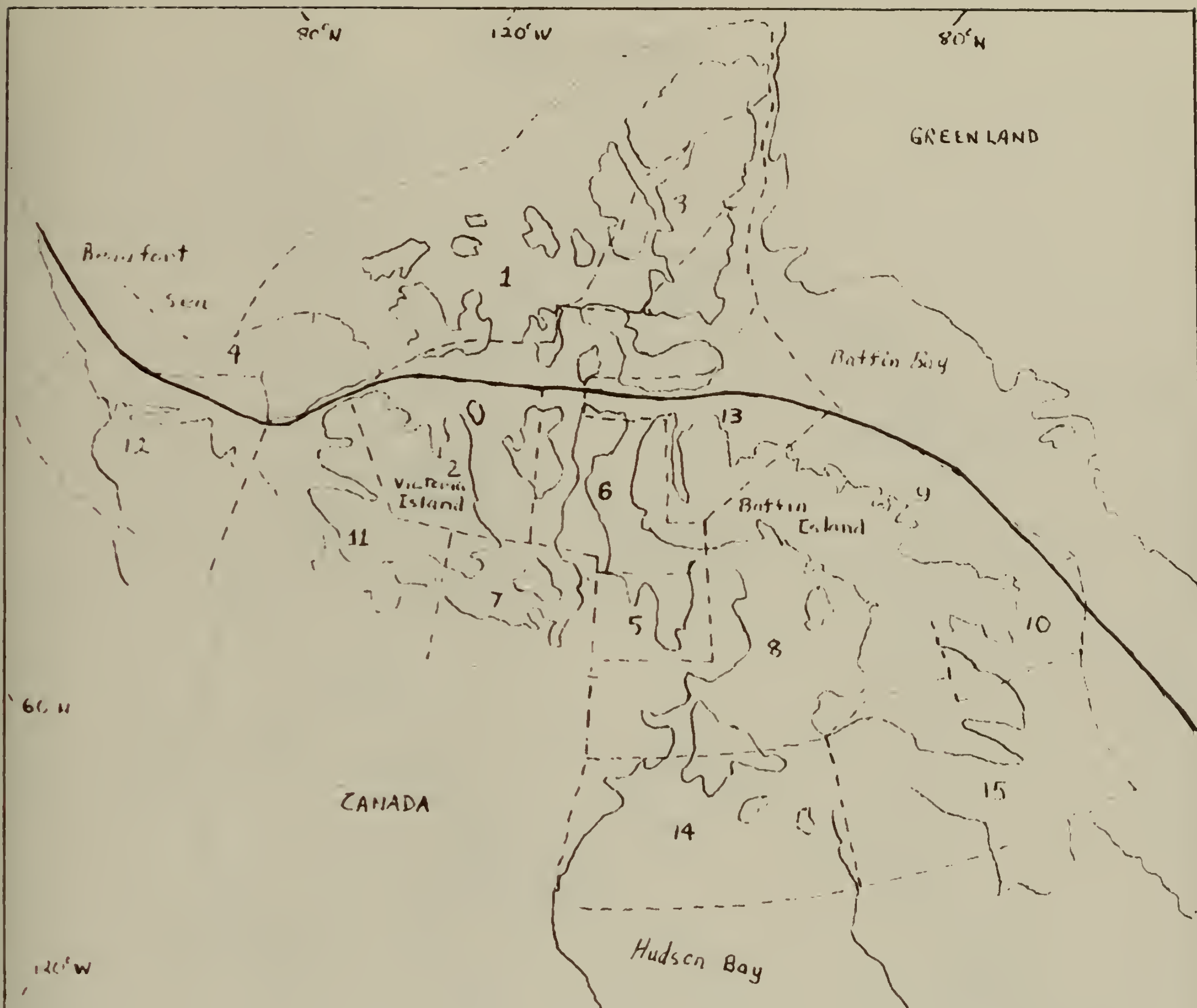


Figure 1.3

Bering Strait

Table 1.2

SEASONAL OPERATING CONSTRAINTS

Northwest Passage

	Vessel Class		
Zones	9	6	3
12	Yr. Round	Yr. Round	Mid June-Jan.
11	Yr. Round	July-March	July-Mid Dec.
2	Yr. Round	Aug.-Oct.	Mid Aug.-Mid Sept.
6	Yr. Round	Mid July-Feb.	Aug.-Nov.
13	Yr. Round	Yr. Round	Mid June-Dec.
9	Yr. Round	Yr. Round	Mid July-Mid Jan.

Bering Strait Route

	Vessel Class		
Zones	9	6	3
4	Yr. Round	Mid July-Nov.	Mid July-Oct.
2	Yr. Round	Aug.-Oct.	Mid Aug.-Mid Sept.
6	Yr. Round	Mid July-Feb.	Aug.-Nov.
7	Yr. Round	July-March	Mid July-Mid Dec.
8	Yr. Round	July-March	Mid July-Dec.
10	Yr. Round	Yr. Round	Mid July-Mid Jan
13	Yr. Round	Yr. Round	Mid June-Dec.
14	Yr. Round	Yr. Round	Mid June-Mid Jan.

three icebreaker classes. Class 9 vessels can operate year round in either route. Class 6 vessels are limited to operation in zones 6 and 2 (Northwest Passage) from mid-July to October (Bering Strait). Zone 2 further restricts Class 3 vessel operations from mid-August to mid-September (30 days).

Chapter I has established the relevant mission requirements and defined the operating region, including seasonal limitations. Chapter II will investigate methods for determining a ship's ice resistance. Powering necessary to achieve the required icebreaking capability will then be established.

CHAPTER II

POWER DETERMINATION

This chapter will investigate current methods of predicting ice resistance for ships (i.e. empirical, semi-empirical, and theoretical). A method will be selected and validated utilizing the POLAR STAR hull form as a reference ship. This technique will be utilized in ship system development in Chapters III and IV. The minimum power necessary for the POLAR STAR hull form to meet the icebreaking requirements will also be determined. This will provide an approximation for SHP inputs for the models to be developed in succeeding chapters.

Determining the power requirements for a ship is a complex process. Analyzing the powering needs of an icebreaker is even more difficult. An icebreaker must be designed to operate effectively in the open water, but more importantly, in the ice. The ship-ice interface introduces many complexities, some of which have eluded adequate theoretical description. There are many characteristics which enter the resistance problem. Many of these are the standard ship's characteristics with which naval architects have an intimate familiarity. In addition, the characteristics of the complex structural material ice must be considered.

Ice is an enigma in itself. Its lack of homogeneity is most aggravating to the engineer. Vance [16] describes ice as a "visco-elastic plastic crystalline material." The mechanical and physical properties of ice are a function of its salinity, temperature, and temperature history. The thickness and growth rate vary widely, depending on the frequency of pressure ridges, local currents, wind, etc.

The important ice parameters, in addition to thickness, affecting the ship-ice interface are density, flexural strength, coefficient of friction, elastic modulus, Poisson's ratio, and the compressive strength. Obviously, these parameters are beyond the control of the ship designer. The Canadian zonal classification system is a valuable attempt to provide naval architects with useable ice condition guidelines. The ship parameters are very much under the designer's control. In order to intelligently select these parameters, the designer must be aware of the icebreaking phenomenon.

An understanding of icebreaker design must begin with an introduction to the failure mechanism of ice. As the ship traverses open water infiltrated with broken ice pieces, it impacts a solid ice sheet. This applies a horizontal load to the edge of the ice sheet, causing the ice sheet to be placed in a state of plane stress. If the load is sufficiently large (or the sheet thin enough), the

ice will crack and fail perpendicular to the edge of the sheet. Given that the ice is not constrained, the crack will be forced open and wedged apart by the ship's stem, allowing the vessel to continue forward [12]. This is the most elementary form of the continuous mode of icebreaking.

The more complex mode of uninterrupted icebreaking occurs when the solid ice sheet is too thick for the initial impact to generate a large enough crack. As the vessel impacts with the ice, radial cracks will be formed. Local crushing of the ice at the bow will also take place, but the ice will not fail. When the "applied force per unit face area" in the notch "equals the compressive strength of the ice," crushing will cease and the vessel will ride up onto the ice [8,9]. The vessel rides up onto the ice as a result of forward velocity and an inclined bow. Kinetic energy is lost, small changes in draft and trim take place, and a vertical force is exerted on the ice sheet. This vertical force places the sheet in bending, causing radial cracking to extend and multiply in number. At some point a circumferential crack will be formed, the ice fails, and ice wedges will be broken off (Figure 2.1). The vessel will then fall, displacing the wedges down and away from the ship [8,9]. If the vessel retains some of the forward velocity and accelerates to the next impact, having no net kinetic energy loss, continuous icebreaking is being accomplished.

Figure 2.1

Crack Patterns

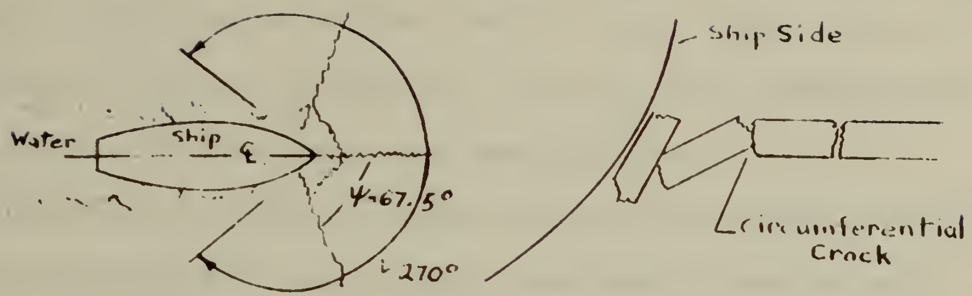


Figure 2.2

White Bow

The limiting case of the uninterrupted mode appears when all of the vessel's kinetic energy is utilized to break the ice. Obviously, the changes in draft and trim are larger, and the vessel retains no residual forward velocity after falling and displacing the broken wedges. The ship must then withdraw from the ice, back off, and accelerate into the ice sheet. Intuitively, ramming is not a very efficient mode of breaking ice.

Breaking ice with a ship form can be accomplished in the two basic modes previously discussed, ramming and continuous. Each will be dealt with in more detail in following sections. There is no specific design constraint contained in reference [1] for thickness of ice that can be successfully broken in the ramming mode. However, a discussion is included due to the significance of ramming in the vessel's ice capabilities, primarily related to pressure ridges, and in the interest of completeness.

2.1 Resistance--ramming mode

The idea for analytical description of the icebreaking phenomenon is a fairly recent concept. In 1888, R. Runeberg constructed the first equation, recognizing the importance of the vertical component of force exerted by the vessel onto the ice [10]. Many learned men followed, with White [10] providing the most recent in-depth theoretical analysis of the icebreaker bow form.

The purpose of White's research was to develop an equation to predict "the dynamically developed force at the bow of an icebreaker during encounter with virtually unyielding ice," and subsequently, to identify parameters which the designer could utilize to form a more efficient bow [10]. White splits icebreaking into several states and phases similar to those previously described. White then formulates equations for each state or phase, and establishes a computer program to simultaneously solve these equations. The program is contained in reference [10]. A more thorough discussion is contained in Appendix A. In light of present technology, White's program could be considered rather basic.

An important output of White's research was the bow form which was recommended for optimizing ramming efficiency (Figure 2.2). A modified version of the White bow was utilized on the U.S.S. MANHATTAN, U.S.C.G.C. POLAR STAR, and U.S.C.G.C. POLAR SEA.

White maintains that an icebreaker should be judged by its "ability to impart a relatively sustained force to the ice in the vertical direction" [10]. The ability of a vessel to climb onto the ice and exert a downward force is strongly dependent on displacement, impact velocity, bow angle, and the spread angle complement. Based on Kashteljan [13], Lewis and Edwards [14,15], Vance [16], and Milano [8,9], it appears that although

these variables are important: beam, installed horsepower, ice thickness, and ice flexural strength play a much more significant role. Continuous icebreaking capability must, in addition to the failure of an ice sheet, address the total resistance of the ship in the ice.

2.2 Resistance--continuous mode

In the past decade a great deal of effort has gone into the study of icebreaker resistance in the uninterrupted mode. One of the useful outputs of knowing the resistance, is the determination of the necessary power to break a given ice thickness. The Canadians have addressed this question of installed horsepower in their Arctic Classification Rules.

2.2.1 Empirical resistance

In order to operate a vessel in Canadian waters, the vessel must meet the power requirements commensurate with its Arctic Class, zone of operation, and season of transit. The zones and Arctic Classes are contained in Chapter I. Table 2.1 contains the necessary coefficients and equations to determine the power requirements for a vessel based on ice class and open water speed (4). Utilizing the POLAR STAR parameters as inputs, the minimum required power, calculated in Table 2.1, is 67,242 SHP. This equation has been empirically developed utilizing icebreaker data for vessels through Arctic Class 4. The

Table 2.1

REQUIRED MINIMUM SHP
FOR
ARCTIC SHIPS

Arctic Class	Minimum Speed in Knots	Value of A
1	16	1
1A	18	1.5
2	20	2
3	22	3
4	22	4
6	24	6
7	26	7
8	26	8
10	26	10

Assumes minimum speed in ice of 3 knots.
The required shaft horsepower shall not be less than the horsepower calculated from the following formula:

$$\text{Required SHP} = Pr * \frac{Dr}{D} \text{ if } \frac{Dr}{D} \geq 1; \text{ or } = Pr \text{ if } \frac{Dr}{D} < 1$$

Where $Pr = (22-0.13W^{1.3})BA^2$ foot pound system

W = displacement of the ship in tons
B = maximum breadth of the ship in feet
A = the value of A for an arctic ship given in the above table

$$Dr = 4.5\left[\frac{Pr}{1000Z}\right]^{1/2}$$

Z = the number of propellers
D = the diameter of propellers in feet

POLAR STAR

W = 10,863 tons
B = 78 feet
A = 6
Z = 3
D = 16 feet

$$Pr = (22-.1(22.15)78(36) = 55{,}557.1 \text{ SHP}$$

$$Dr = 4.5\left[\frac{55{,}557.1}{1000(3)}\right]^{1/2} = 19.36$$

$$\frac{Dr}{D} = \frac{19.36}{16} = 1.21 \quad \text{Required SHP} = Pr * \frac{Dr}{D} = 67{,}242 \text{ SHP}$$

applicability to the higher Arctic Classes remains to be validated. It is felt that experience in future years will indicate a lower power requirement for the higher classes. Further discussion later in this chapter will demonstrate that a significantly lower power level is needed to enable the POLAR STAR to break six feet of ice in the continuous mode.

The Canadian powering equation is of limited value to the designer. It seems to overestimate and it assumes that a great deal of knowledge about the vessel is known. The equation was developed not for use as a design tool, but primarily as a safety criterion.

In addition to the empirical derivation developed by the Canadians, progress has been made in a more theoretical vein. This work has taken two principal forms, semi-empirical and pure theoretical.

2.2.2 Semi-empirical resistance

Possibly the "most significant contribution toward a real understanding of the continuous icebreaking mechanism" [8] was developed by Kasteljan et al. [13], in 1968. The expression for total ship's resistance is a summation of, not only resistance due to failure of the ice sheet, but also, resistance due to momentum exchange between the ship and ice, ice buoyancy forces, and open water resistance. The equation takes the following form:

$$R(\text{total}) = K_1\gamma_i Bh^2\mu_o + K_2Bh\sigma_i\mu_o + K_3B^xhV^y\left(\frac{1}{\eta^2}\right) + R(\text{open water})$$

where:

K_1, K_2, K_3, and exponents x and y must be determined from model and full-scale data.

(K_1 = 3.6, K_2 = 0.004, K_3 = .25, x = 1.65, Y = 1 for ERMAK)

B - beam at waterline

h - ice thickness

V - ship's velocity

γ_i - specific weight of ice

σ_i - ice flexural strength

μ_o - force coefficient related to hull shape, expressing the efficiency with which horizontal thrust is converted to vertical force on the ice.

η - force coefficient related to pushing broken ice away, expressing the efficiency of conversion of horizontal force to an athwartships component.

Kashteljan based this work on data from the icebreaker ERMAK. An investigation of the equation brings to light several interesting points. The velocity dependent term is linear with velocity, which appears to contradict full scale results. The equation is not dimensionally consistant (i.e. K_1 and K_2 are dimensionless, but K_3 must

be dimensional). Kashteljan's equation does not address length or draft. Nor does it explicitly include the coefficient of friction.

Lewis and Edwards [14], in 1970, made a contribution to the field by developing an equation utilizing regression analysis techniques. The equation had a form similar to Kashteljan's:

$$R_{(ICE)} = C_1\rho_i gh^2 + C_2\sigma_f h^2 + C_3\rho_i BhV^2$$

where:

C_1, C_2, and C_3 are results of regression

ρ_i - mass density of ice

g - acceleration of gravity

h - ice thickness

B - ship's beam at water line

σ_f - ice flexural strength

V - ship's velocity

Model and full-scale data are needed to determine the coefficients.

Length, draft, and the coefficient of friction are not included in the equation, nor in treating the data. Vance [16] points out that the equation is nondimensionalized utilizing the strength term prediction variables, $\sigma_f h^2$, "which although formally correct, leaves much to be desired because of the variability in the control of σ_f and E."

Edwards, et al. [15], in 1972, presented a revised resistance formulation based on U.S.C.G.C. MACKINAW data.

$$R_{(ICE)} = C_1 \rho_w g B h^2 + C_2 \sigma_f B h + C_3 \rho_w g^{.5} B h^{1.5} V$$

where:

C_1, C_2, C_3 are results of regression

ρ_w - mass density of water

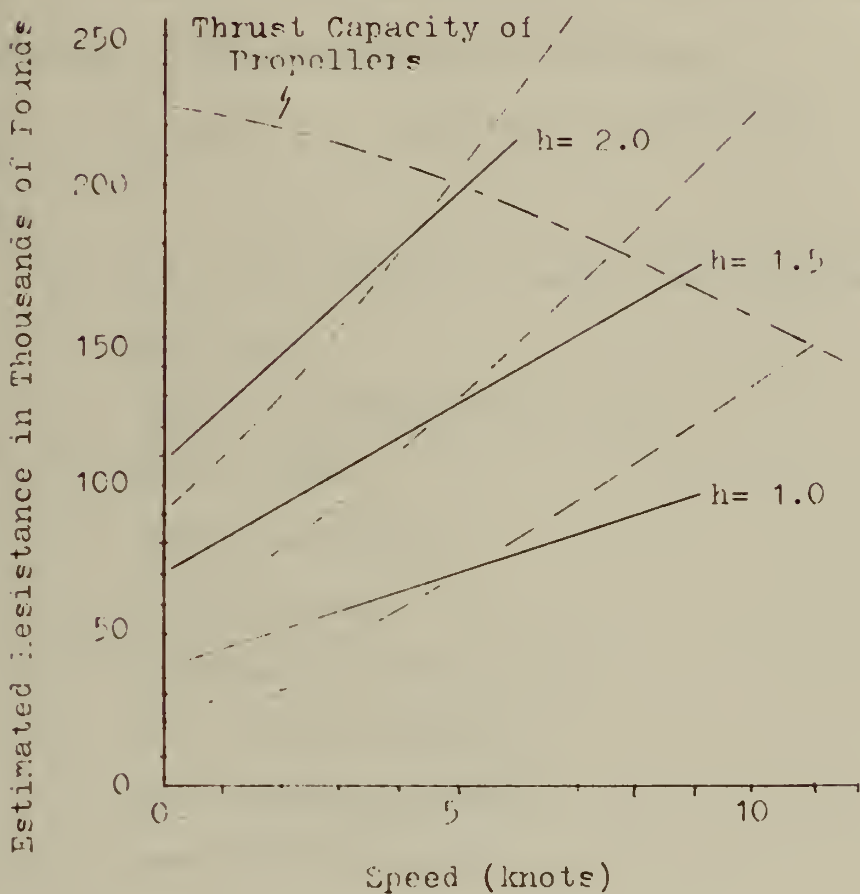
Remaining variables are defined previously.

As in Kashteljan's work, the velocity dependent term has become linear, producing errors at high speeds (Figure 2.3). Length and draft are neglected. However, a sincere effort is made to match the coefficient of friction between the full scale tests and model tests. A new term was selected for nondimensionalizing, the gravity term ($\rho_w g B h^2$), avoiding the controllability problem. Finally, this formulation is for a particular vessel, utilizing model and full scale data. The usefulness of any of these methods to the designer, as a predictor of full scale results based on model data is questionable. This problem was directly attacked by Vance [16].

The final semi-empirical method to be presented is probably the most rigorous contribution to date. Vance has successfully bridged the gap, allowing prediction of full scale ice resistance based on model data alone. Employing dimensional analysis supplemented with stepwise multiple regression analysis, five sets of full scale and model data were utilized in developing an equation for ice resistance. A more in-depth discussion is contained in Appendix B. The investigation sought a resistance

Figure 2.3

Predicted and Measured Full Scale Resistance-Edwards, et al.



Snow cover = 0

Ice strength = 126 lb/in^2

----Full Scale Data

——Model Data

equation with a form common to the five sets of data previously described. Vance states that, "Utilization of the equation developed predicted full scale resistance within 10 percent when sufficient and reliable data was available." The equation is of the form:

$$R_{(ICE)} = C_s \rho_\Delta g B h^2 + C_B \sigma_f B h + C_V \rho_i V^2 L B^{.35} h^{.65}$$

where:

C_S, C_B, C_V are results of model data regression analysis.

L - ship's length at water line

B - ship's beam

V - ship's velocity

h - ice thickness

g - acceleration of gravity

σ_f - ice flexural strength

ρ_w - mass density of water

ρ_i - mass density of ice

$\rho_\Delta = \rho_w - \rho_i$

C_S, C_B, and C_V should be determined from model data, and full scale parameters (i.e. V, h, B, L, σ_f, etc.), should be inserted into the equation to predict full scale resistance. The equation is "reliable for large icebreakers." Vance mentions limited applications for shorter, high beam-length ratio ships, bearing in mind a necessary modification of the exponents of B and h in the velocity

term. Sufficient data is not presently available to determine their value.

Vance has removed the velocity linearity from the velocity dependent term. Draft was considered, but insufficient data ruled out definitive results. The coefficient of dynamic friction was given careful attention. Finally, Vance has included a length variable. This is felt to be of major importance when considering a vessel of tanker size (i.e. U.S.S. MANHATTAN). Generally, if the ice tightness (lateral pressure) is not severe, the length effect on a 400 foot icebreaker is small. An icebreaker is short compared to a tanker (400 vs. 900 feet). The ice will certainly close in around the sides of a tanker, the problem will present a smaller impact on the icebreaker. The tanker also has a large percentage of its length as parallel middle body. An icebreaker has little or no parallel middle body and a maximum beam forward of amidships. The length variable, therefore, becomes more significant as the vessel becomes longer and more wall-sided.

It is clear that one of the major problems with the regression technique is the extreme reliance on data. Semi-empirical methods have been hampered by the lack of good data. With increased awareness, data collection in the future will be more exacting.

The preceding discussion of the development of semi-empirical resistance equations is not meant to be all-inclusive. Only some of the recent studies are mentioned to provide the reader with an idea of current development.

2.2.3 Theoretical resistance

The only pure theoretical approach to icebreaker resistance to date has been proposed by Milano [8,9]. This derivation is based on a total energy balance concept. Milano attacks the icebreaking problem by utilizing an energy balance for the resistive components of each phase of ship motion. Milano assumes a complete cycle of ship motion to include:

a. Motion through ice-filled water to impact;
b. Ramming or impact with the unbroken pack;
c. Crushing of the ice;
d. Climbing onto the ice;
e. Falling of the ship.

Resistive components overlap from one phase to another. Resistance due to travel in ice-filled waters "acts continuously and concurrently" with resistance components from the other phases [8]. Ramming and crushing take place so close together in time that they can be grouped together for resistance considerations. Climbing or sliding onto the ice "is an independent mechanism that involves a transfer

of kinetic to potential energy plus significant friction forces" [8]. The falling of the ship is also an independent mechanism, which will include resistive components associated with inertia, buoyancy, and friction. The available energy in the system is the sum of kinetic energy of the vessel at impact and the propeller thrust input over the distance of the complete cycle. The total energy expended is the sum of the energies involved in each event of the cycle. Therefore, "the resistance to ship motion can be approximated as that average force acting through the cycle distance which produces the given energy level" [8]. The total resistance (F_T) to ship motion is the total energy lost (E_T) divided by the complete cycle distance (x').

$$F_T = \frac{E_T}{X'}$$

where

$E_T = E_1 + E_2 + E_3 + E_4 + E_5$

E_1 = energy of ship motion through broken ice

E_2 = energy of impact of ship with the unbroken ice field

E_3 = energy of ship motion onto the ice field

E_4 = energy of ship motion after ice failure

E_5 = energy involved in the ice slab rotation and submersion

A scenario for thick ice, the most severe resistance condition, is presented by Milano [8]: the ship will move through the ice-filled channel (E_1), impact the ice causing loss of all available impact energy and local crushing (E_2), climb onto the ice until ice failure (E_4) and move forward, forcing the ice down and out of the way (E_5). A more detailed description of the energy components and a discussion of thinner ice conditions are contained in Appendix C.

It would seem reasonable to expect some compatibility between Milano's and White's analytical results. Since their descriptions of icebreaking phases are similar, it would appear that ramming, as White describes it, is the limiting case for Milano. In "Variation of Ship/Ice Parameters on Ship Resistance to Continuous Motion in Ice" [9], Milano suggests that as ship's speed of advance approaches zero that E_1, E_2, and E_5 tend to zero. This leaves E_3, loss due to climbing onto the ice, and E_4, the ship's falling after ice failure. White does not address the situation after ice failure, i.e. the ship falling and submerging the broken ice pieces. Therefore, any attempt to compare the two analytical models must only address the ship's resistance due to the E_3 energy loss. Figure 2.4 indicates that the U.S.C.G.C. MACKINAW at zero speed (E_3 alone) could break 4.8 feet by Milano and 4.55 feet of ice based on White. This is reasonable compatability.

Figure 2.4

Ramming - Milano and White

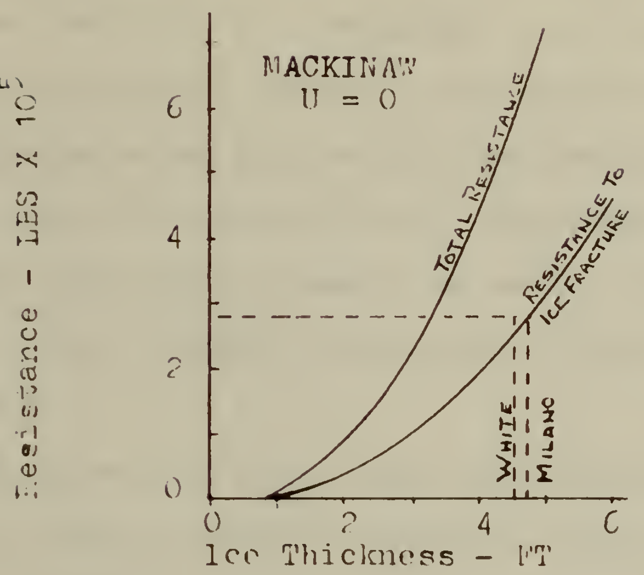
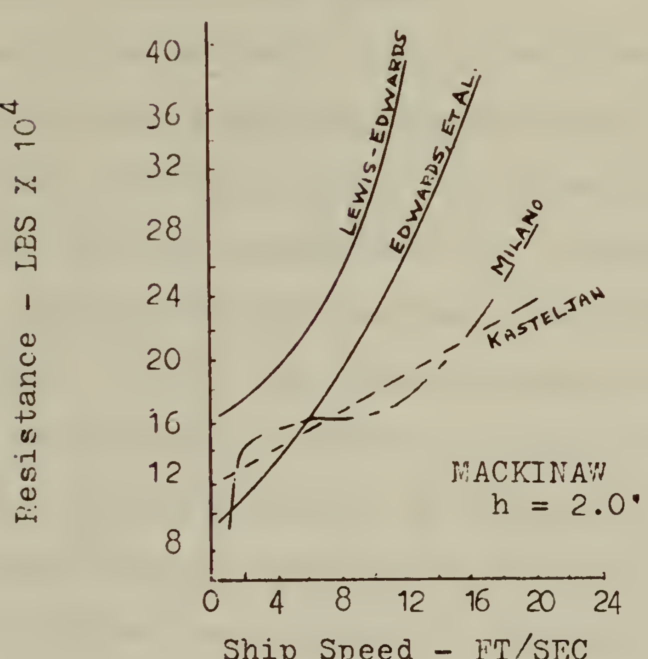


Figure 2.6

Resistance Predictions Compared



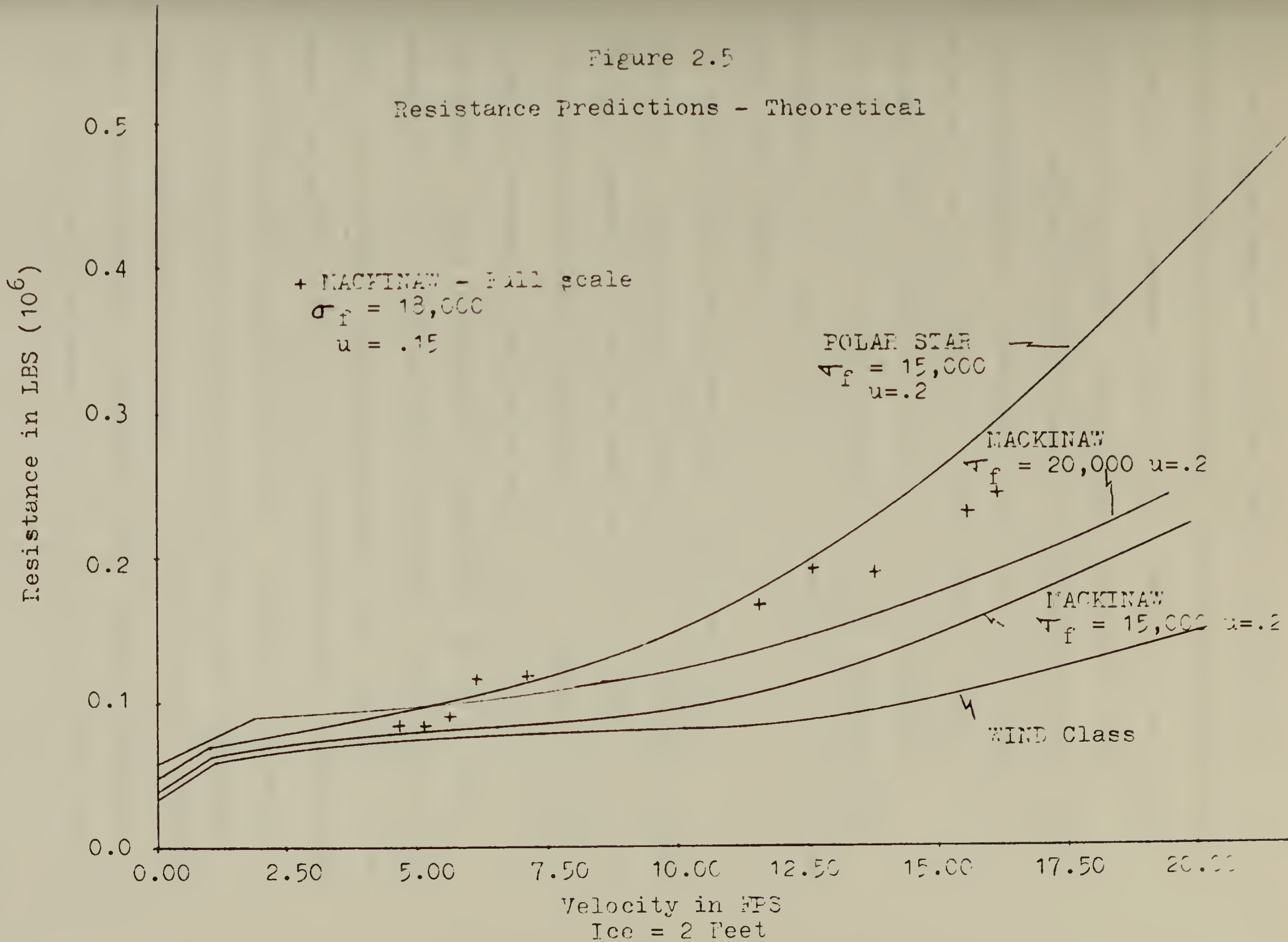
The equations derived by Milano [8] were formed into a computer program. Initially, the program was run with inputs from the U.S.C.G.C. MACKINAW, U.S.C.G.C. STATEN ISLAND, and U.S.C.G.C. RARITAN. The outputs were compared with full-scale data; good correlation was found [8]. Two years later [9] the U.S.C.G.C. MACKINAW was utilized as the base vessel for a vessel parameter and ice properties study, further discussion is contained in reference [9].

In the summer of 1975, Vance modified Milano's program (primarily, output format), formed data input bases for several icebreakers, and did extensive work with the POLAR STAR [18]. The POLAR STAR was commissioned in January 1976. The correlation of full-scale and model results should be very interesting indeed.

The work done by Vance [18] illustrates further validity for Milano's program (Figure 2.5). The best data available from full-scale tests are presently felt to be that of the U.S.C.G.C. MACKINAW [17]. Milano's predicted resistance curve correlates very well with the U.S.C.G.C. MACKINAW data. Figure 2.6 displays Milano's resistance prediction in relation to several of the semi-empirical methods [9]. Milano's program is considered to be the best icebreaker design tool available today [17]. Some of the limitations are presented in Appendix C.

Figure 2.5

Resistance Predictions - Theoretical



Milano's computer program, with its myriad of necessary inputs (see Appendix C), is not a trivial tool to employ. It assumes that a great number of the ship's parameters are known. The designer can utilize this program, making approximations of secondary elements, and running systematic parameter studies for optimization of resistance and icebreaking characteristics. Milano presents such a study for the U.S.C.G.C. MACKINAW in reference [9]. A modified version of this approach will be employed in a later section to investigate various sized POLAR STAR geosims.

In anticipation of merchant requirements for resistance calculations on vessels with high arctic classification numbers, a warning must be given. The semi-empirical models presented in 2.2.2 assume that the Reynolds boundary layer resistance component can be neglected. This is generally considered to be a valid assumption for a shallow draft icebreaker operating in the ice at low speeds, three to six knots. The boundary layer cannot establish itself in the above situation because the ice continually disrupts the layer. However, an Arctic Class 8, deep draft (greater than thirty feet) merchant vessel, operating at 16 knots in two feet of ice, probably has a boundary layer. Therefore, the Reynolds boundary layer resistance component will be present in some form [17].

What happens to the models for predicting resistance? This presents a very interesting question for further research.

2.3 Shaft horsepower determination

The mission requirements for the original escort/ logistic support/scientific icebreaker include an open water speed of 17 knots and a six foot continuous icebreaking capability. This section attempts to determine the SHP necessary to meet these requirements. The POLAR STAR was initially used as a base ship. This assumes the icebreaking hull form is efficient, and the vessel contains the minimum volume and displacement necessary to perform its multifaceted mission. The POLAR STAR was also used because of the existing data base on computer disks developed by Vance for Milano's program. Several steps or phases will be utilized to accomplish the goal: investigate past designs, employ Milano's computer program, check feasibility with Vance's resistance equation, and evaluate the impact on SHP of snow cover and ridging.

An investigation of past designs carried out by the author resulted in the Δ/SHP and B/SHP curves of Figures 2.7 and 2.8. Beam was selected, being the ship's characteristic most explicitly impacting the resistance described by the semi-empirical equations. These curves, based on

Figure 2.7

Δ vs. SHP

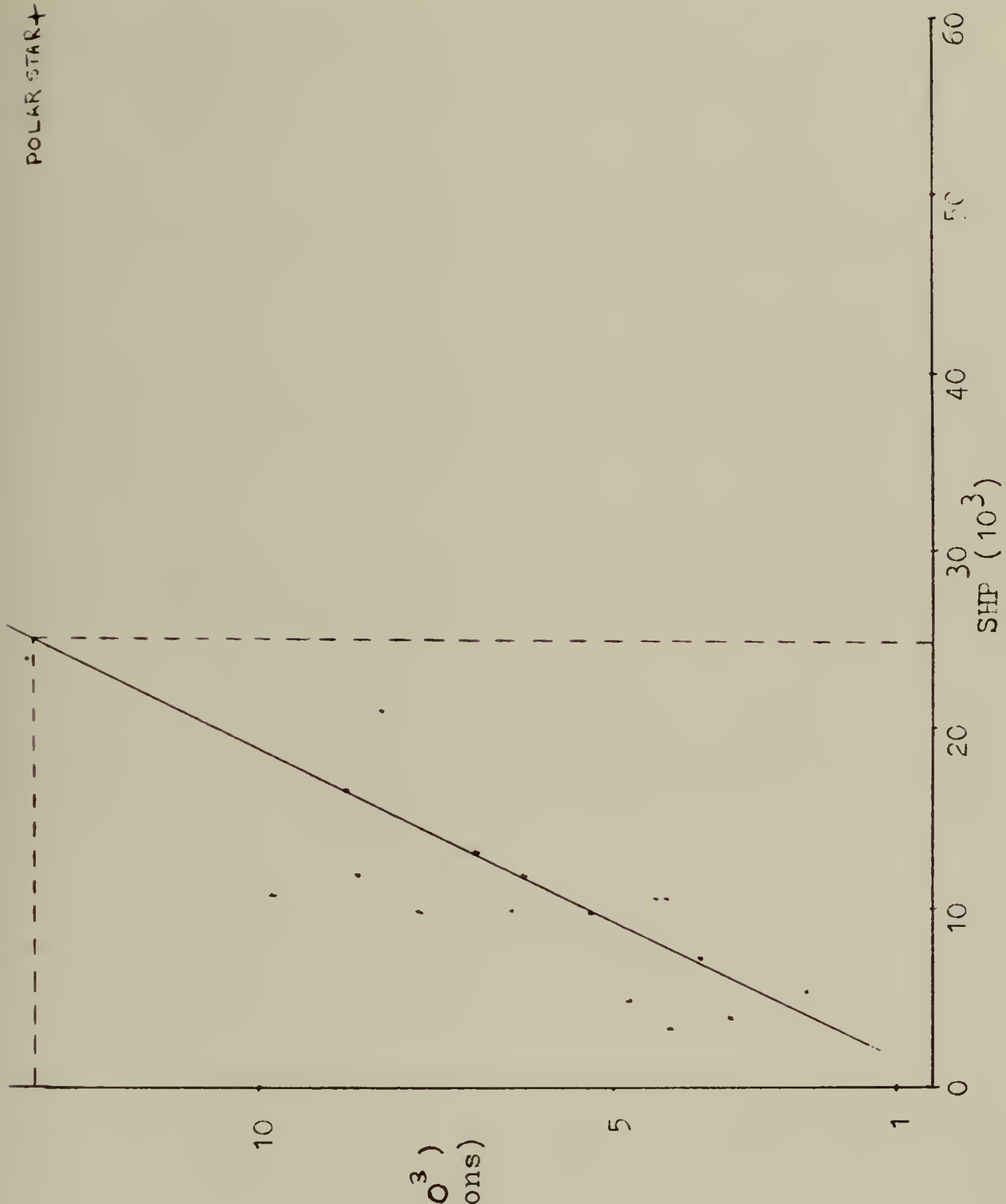
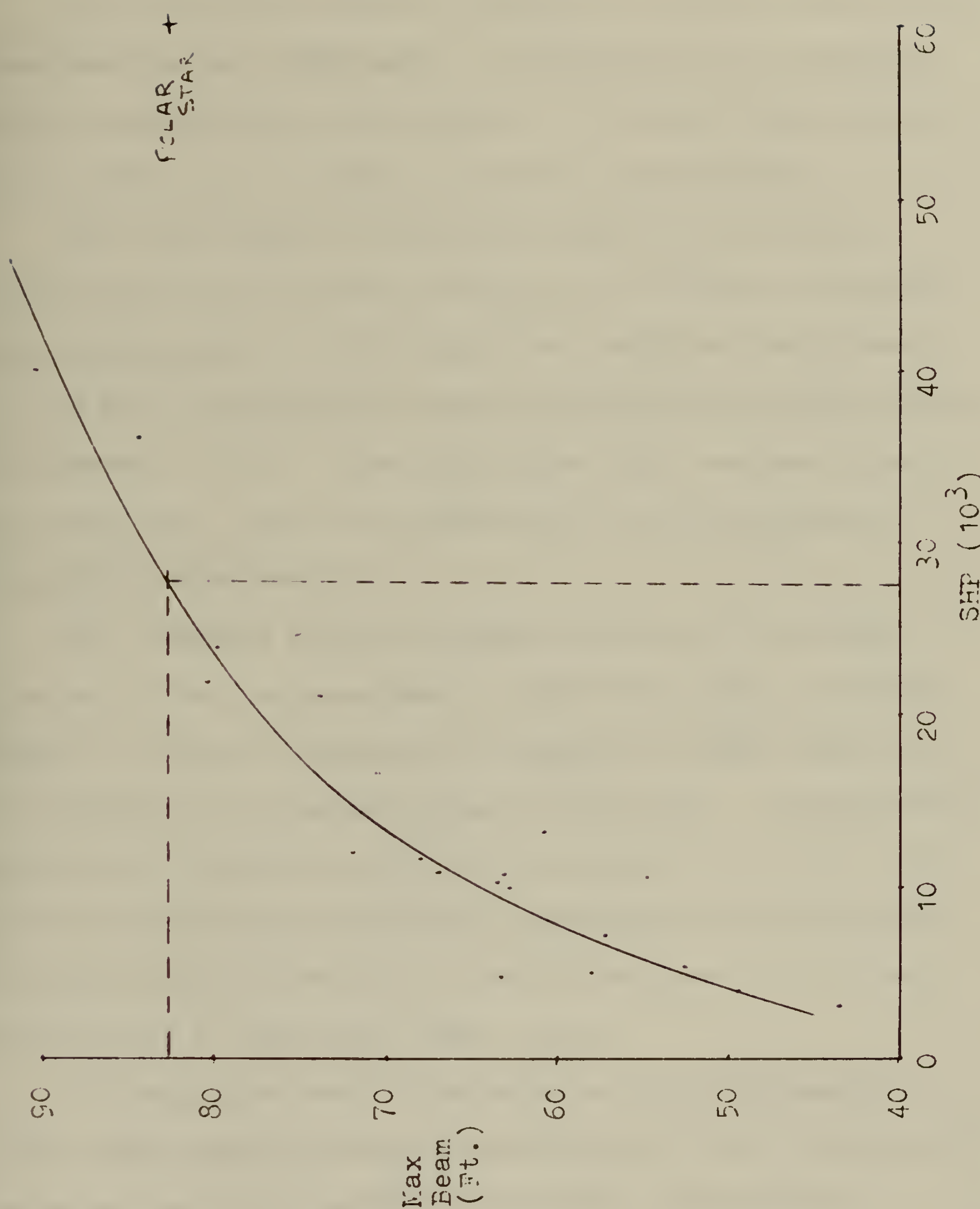


Figure 2.8

Beam vs. SHP



data contained in Appendix D, indicate that a vessel with a displacement of 13,190 tons would have 25,000 to 30,000 SHP.

At this point a direct analysis of each of the design requirements was undertaken. Investigation of the POLAR STAR's speed power curve (Figure 2.9) showed that 18,000 SHP is required to attain 17 knots in open water.

The requirement to break six feet of ice in the continuous mode required utilization of Milano's program. The program used in this thesis was modified and placed on the Navy's Underwater Sound Lab computer by Vance during the summer of 1975. The POLAR STAR data base was created at that time. The data comprising this input base are displayed in Appendix D.

The computer program outputs resistance in pounds. This must then be converted to installed shaft horsepower. Table 2.2 shows a reduction of computer output indicating 29,400 SHP as the desired power capability. An open-water propulsive coefficient of 0.6 was reduced to 0.5 to allow for ice interruption of water flow around the propellers. The more severe case of σ_f = 20,000 psf and μ = 0.3 was selected as a reasonable upper limit.

To check the overall validity of the resistance, a rough check was performed using Vance's (1974) resistance equation, with the MACKINAW full-scale coefficients.

Figure 2.9

POLAR STAR Speed/Power Curve (Estimated)

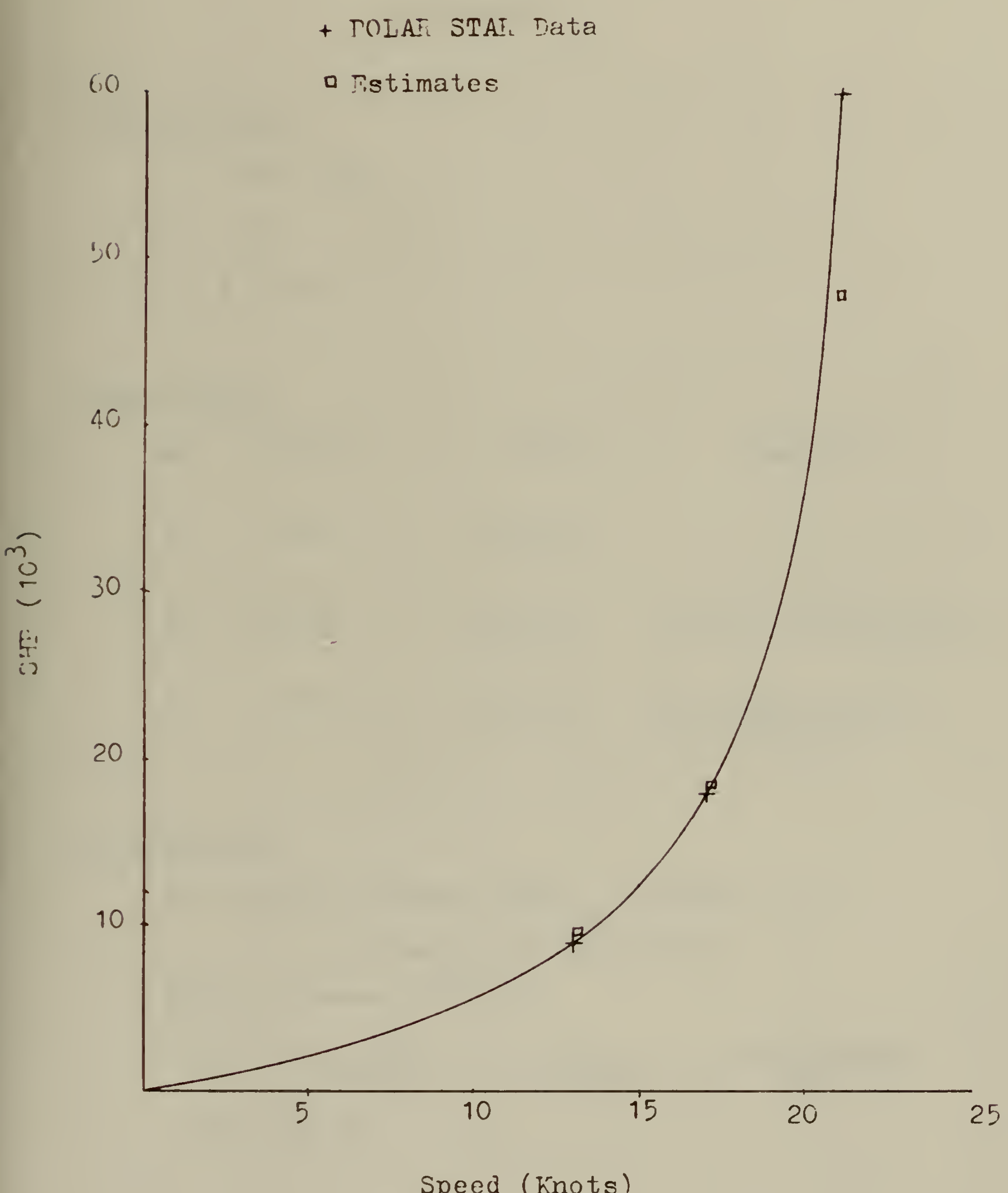


Table 2.2

SHAFT HORSEPOWER

DETERMINATION

Fixed Parameters

σ_c = 50,000 psf

S.G. = .92

V = 6 ft./sec.

Computer Runs

Run	σ_f(psf)	μ	R(lbs)	Change (%)
15	15,000	.2	570,000	
30	20,000	.2	860,915	51 due to 5000 increase in $_f$
22	15,000	.3	686,000	20.4 over run 15 due to increase in μ

SHP Calculation

Pick the most stringent case, σ_f = 20,000, μ = .3

$$R = 1.204\ R_{30} = 1{,}036{,}541 \text{ lbs.}$$

Service allowance (SA) = 1.3
PC = 0.5

$$SHP = SA\ \frac{EHP}{550\ (PC)} = 1.3\ \frac{RV}{550\ (PC)} = 1.3\ \frac{(1{,}036{,}541)6}{550\ (.5)}$$

SHP = 29,400

The MACKINAW was selected because it was shown to be the most efficient of the vessels investigated by Vance. It must be pointed out that the MACKINAW's bow propeller may have partially accounted for this. However, if progress has taken place, the POLAR STAR should be at least as effective as the much older MACKINAW. These calculations are presented in Table 2.3. The close agreement between methods indicates validity, at least, for conceptual design.

The reader must be reminded that neither Milano nor Vance account for snow cover or ridging. There are several qualitative discussions in the literature attempting to deal with snow cover. The lack of data and extreme difficulty in modeling this phenomenon have prevented definitive resolution of the problem. Edwards et al. [15] propose that each inch of snow cover adds about two tons of resistance to the total. These results are based on data obtained in fresh water tests (Great Lakes) for the MACKINAW. Employing the same procedure as in Table 2.2, each inch of snow is equivalent to, approximately, 130 SHP. To account for this possible snow cover effect, the power capability was increased to 30,000 SHP.

One final effect was considered in the power selection, pressure ridging. There is also little quantitative data available dealing with this problem. The first area of difficulty encountered when discussing ridging is the

Table 2.3

RESISTANCE CALCULATION

USING VANCE

MACKINAW Coefficients (Full Scale)

$C_S = 16.91$

$C_B = 0.034$

$C_V = 0.165$

$\rho_i = .92 \quad \rho w = .92(1.9399 \text{ lb sec}^2/\text{ft}^4) = 1.7847 \text{ lb sec}^2/\text{ft}^4$

$$R_{(ice)} = C_S\rho_\Delta gBh^2 + C_B\sigma_f Bh + C_V\rho_i V^2 LH^{.65}B^{.35}$$

$$= 16.91(.1552 \text{ lb sec}^2/\text{ft}^4)32.2 \text{ ft/sec}^2(78\text{ft})\ (6\text{ft})^2$$

$$+ .034(15{,}000 \text{ lb/ft}^2)78\text{ft}\ (6 \text{ ft})$$

$$+ .165(1.7847 \text{ lb sec}^2/\text{ft}^4)\ (6\text{ft/sec})^2\ 352\text{ft}(6\text{ft})^{.65}(78\text{ft})^{.35}$$

$$= 237{,}295 + 238{,}680 + 54{,}945$$

$$= 530{,}920 \text{ lbs.} \quad \text{vs.} \quad 570{,}000 \text{ lbs. by Milano}$$

definition of the ridges in the operating theater. Ridging in nature is a very stochastic process and, therefore, difficult to physically or analytically model.

The primary operating area for vessels in this thesis was defined, in Chapter 1, as the Northwest Passage and the Bering Strait routes to the Prudhoe Bay area. Table 2.4 provides a summary of pertinent data presented by the Arctic Marine Commerce Study [4]. It is apparent that a vessel having only a six-foot continuous mode capability must resort to ramming to overcome these ridges. The reader should be reminded that once the vessel's velocity drops below 3 knots, it is in the ramming mode by definition. The analysis has now shifted to investigating the impact of SHP on ramming capability. The reader must bear in mind that the original mission statement did not contain a specific ramming requirement.

There are two basic issues to be considered when dealing with ramming. The first obvious attribute of importance is the maximum thickness of ice that can be rammed. The vessel being investigated is the POLAR STAR, with only the installed horsepower altered. Investigation of the speed power curve indicates that 30,000 SHP will produce a maximum velocity of 19.5 knots. White [22] shows that the maximum ice thickness broken by ramming is directly proportional to the square root of the ship's velocity. The POLAR STAR can ram 21 feet of ice at 21

Table 2.4

PRESSURE RIDGE OCCURRENCE

Northwest Passage

Zone	August		September		October	
	F	h	F	h	F	h
12	0	0	0	0	5	39
11	0	0	6	42	6	42
2	7	45	7	45	7	45
6	8	45	8	45	8	45
13	0	0	0	0	7	35
9	6.5	30	0	0	6.5	30

Bering Strait Route

Zone	August		September		October	
	F	h	F	h	F	h
4	2	20	2	20	8	20
2	10	25	10	25	9	25
6	2	15	2	15	5	15
7	2	10	2	10	5	10
8	1	5	1	5	2	5
10	1	5	1	5	2	5
13	1	5	1	5	1	5
14	1	5	1	5	1	5

F = frequency in number per nautical mile.

h = thickness in feet.

knots, with 60,000 SHP. The modified ship with 30,000 SHP, having a velocity of 19.5 knots, can ram 20 feet. A review of Table 2.4 shows 20 feet to be just as adequate for the required environment.

Acceleration is the second issue of importance. This relates to the distance from the ice floe over which the vessel must back to attain full speed. Danahy, in the remarks to White [22], proposed a ratio of thrust to ship's weight as an indicator of acceleration potential. Table 2.5 shows that 30,000 SHP provides an acceptable ratio.

Therefore, in light of the discussion contained in this section, 30,000 SHP in the POLAR STAR hull represents the minimum solution to the mission requirements.

Table 2.5

THRUST TO WEIGHT RATIO

Thrust for the POLAR STAR from Milano's program at:

V = 0 ft/sec

h = 1 ft

Thrust = 1,004,953 lbs (60,000 SHP)

Thrust for the 30,000 SHP vessel will be half of the 60,000 SHP ship

Thrust = 502,477 lbs (30,000 SHP)

Weight = 10,863 tons (2240 lbs/ton) = 24,333,120 lbs

Thrust/Weight = 2.06×10^{-2}

Ship	TH/WT (10^{-2})
ST. LAURENT	1.53
M-6	1.88 (Proposed vessel similar to POLAR STAR)
LENIN	2.04
POLAR STAR$_{(30K)}$	2.06 (K = 1000)
WIND	2.25
GLACIER	2.35
POLAR STAR$_{(60K)}$	4.10

CHAPTER III

FOSSIL PLANT SELECTION

The minimum power requirements and method of determination necessary to perform the icebreaking mission for POLAR STAR hull form were established in the preceding chapter. The objective of this section is to select the optimum fossil fuel propulsion subsystem and formulate the resulting ship system. To accomplish these goals a multiple step process was undertaken: delineate propulsion plant requirements, establish selection criteria, investigate propulsion options, formulate simple math synthesis model, and check model output validity utilizing Milano's computer program.

3.1 Plant requirements

Establishing propulsion plant requirements for the icebreaker consisted of consulting the original mission statement [1] and current literature dealing with icebreaker propulsion [1,2,5,11,26]. The following list represents the results of the investigation.

--Develop 100% full power torque from zero to full speed both ahead and astern.

--Ability to absorb shock loading from ramming, ice jamming between hull and propeller, and ice striking the propellers.

--Ability to operate effectively under the dynamic load condition imposed by icebreaker motion.

--Maximum reliability.

--Minimum research and development. Only currently available equipment and systems are to be considered.

--Meet the following 77 day operating profile:

10 days at 50% power

45 days at 80% power

7 days at 100% power

15 days drifting at approximately zero power

3.2 Plant selection

There are three types of conventional propulsion plant that have the capability to meet the stringent icebreaker requirements, Diesel Electric (DE), Oil Fired Turbo-Electric (OFTE), and Gas Turbine Electric (GTE) (singly or in combination with diesels, CODOG-E). Preliminary fuel calculations indicate that achieving the 77 day operating profile will necessitate a ship larger than the POLAR STAR. This results from the additional displacement necessary to balance the increased fuel weight. Generally, as the ship expands in size, the costs of materials, labor, maintenance, etc. also increase. Consequently, the selection criteria will direct the search toward the conventional plant that minimizes the propulsion subsystem impact.

As a result of the operating profile, the specific fuel consumption (SFC) becomes a primary propulsion plant parameter. Table 3.1 lists specific fuel consumption rates, from various sources, for the plants of interest to this study. It must be pointed out that the values presented in reference [11] were based specifically on icebreaker applications. This reference also displays the fuel rates as a function of the percent of total horsepower utilized (Figures 3.1, 3.2, and 3.3). This particular form of presentation lends itself well to an operating profile analysis. The values for SFC given by the NUS Corporation [11] appear to be high relative to the others. However, they are not unreasonable and do apply directly to icebreakers. The others provide guidance for the general case. Therefore, the plots contained in Figures 3.1, 3.2, and 3.3 will be used to perform fuel weight calculations.

Propulsion plant weight is another important parameter. Several estimating techniques are discussed briefly in the ensuing paragraphs. Table 3.2 contains a survey of machinery weights. Due to the lack of compatibility between authors, the table contains only relative weights or trends of machinery types as shown by each author.

Marine Engineering [27] provides data in graphical form, with the specific weight (the weight of the complete

Table 3.1

SPECIFIC FUEL CONSUMPTION

Plants	References			
	M.E.	HEW.	FEM.	NUS
Diesel Electric	.43			.49
Diesel (Medium Speed)	.41	.40	.46	
OFTE	.57			.68
Steam	.52	.51	.58	
GTE (Waste Heat)				.65
Gas Turbine	.50	.53	.60	

Notes

Specific fuel consumption rates are all purpose.

M.E. - Marine Engineering [27]

HEW - Hewitt [28]

FEM - Femenia [29]

NUS - [11]

Figure 3.1

Overall Fuel Consumption

Diesel

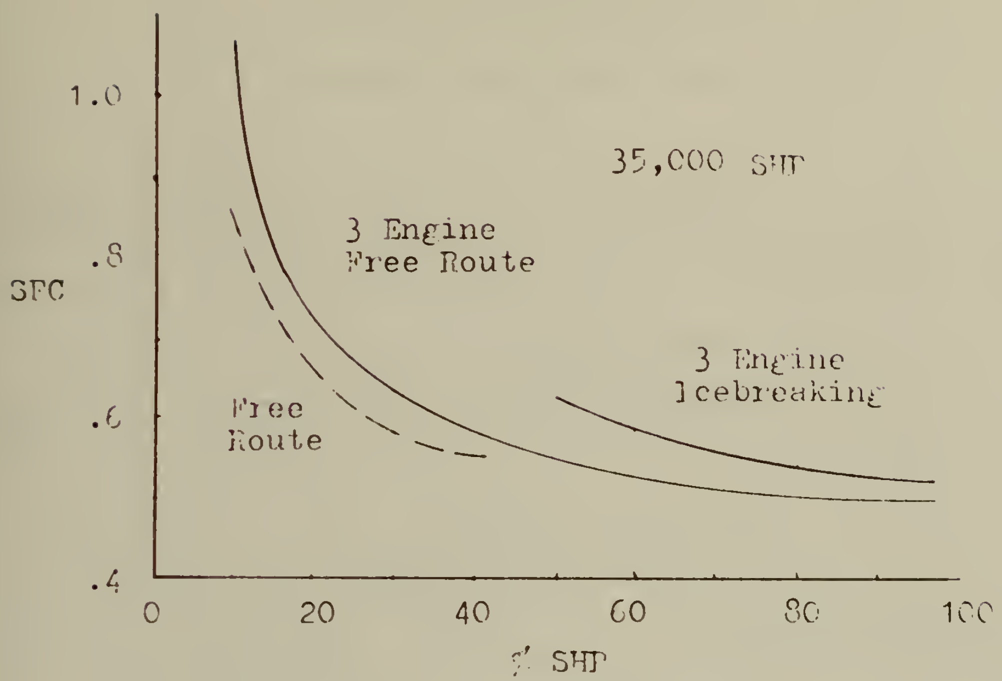


Figure 3.2

Oil Fired Steam Turbine

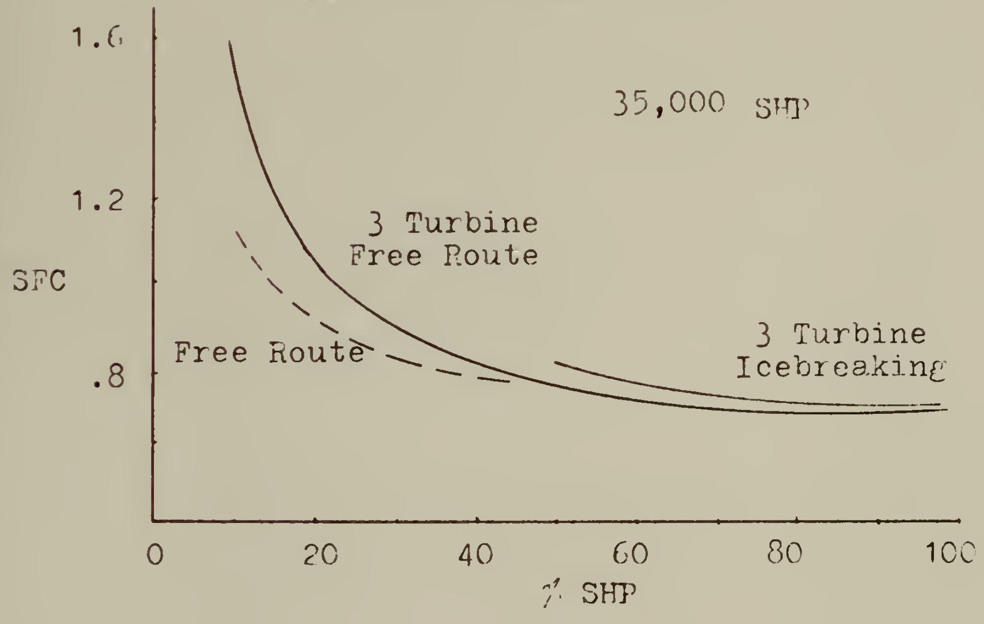


Figure 3.3

Overall Fuel Consumption

Gas Turbine Utilizing Waste Heat

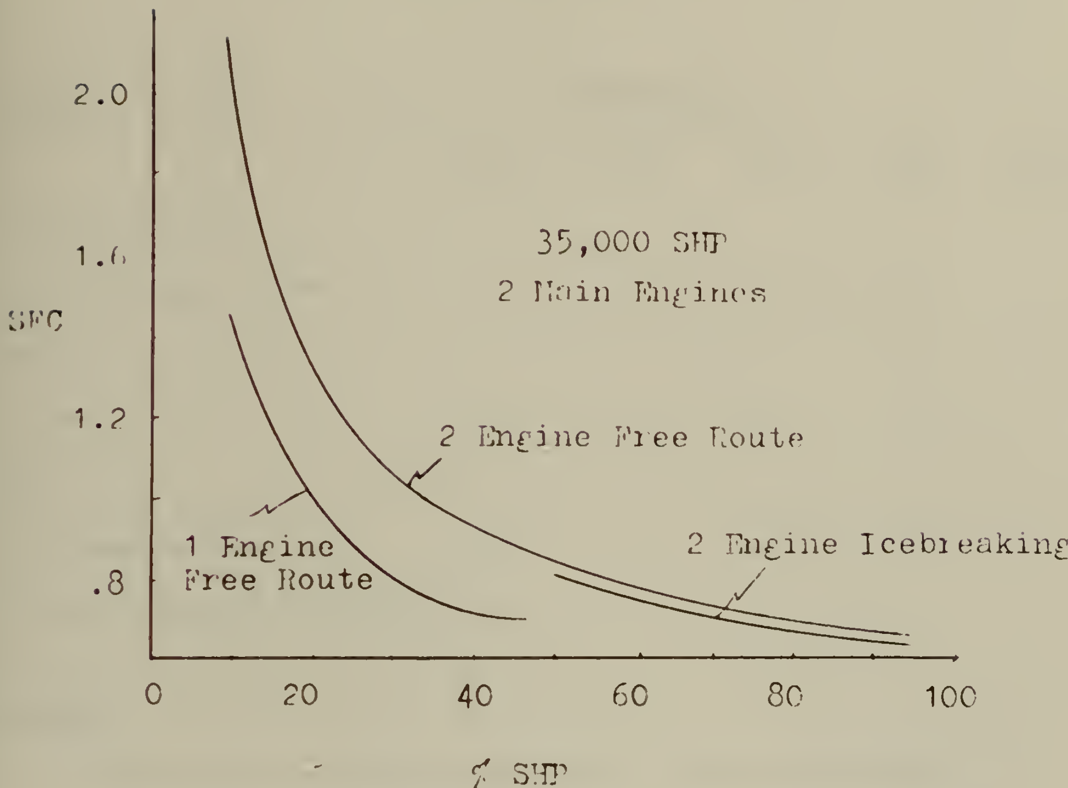


Table 3.2

RELATIVE WEIGHT OF

PROPULSION PLANTS

Plants	References				
	M.E.	HEW.	F&S	F&M	NUS
Diesel Electric					2
Diesel (Medium Speed)	3	3	2	2	
OFTE					3
Steam	4	2	3	3	
GTE					1
GT (Regenerative)	1	1	1	1	
CODOG (Implied Regenerative)	2				

Notes

Plants are rated from 1 to 4 with 4 being the heaviest.

The addition of the Electric System causes increases above the systems' standard weight.

M.E. - Marine Engineering [27]

HEW. - Hewitt[28]

F&S - Frankel and Simpson [31]

F&M - Frankel and Marcus [30]

NUS - [11]

propulsion plant per unit of rated shaft horsepower) plotted against shaft horsepower. The data are "representative propulsion plant weights." No data are presented for combined Electric Systems and no indication of the number of shafts is given. It is not clear but can be inferred that the CODOG plant includes a regenerative gas turbine.

Hewitt [28] presents graphical data with power plant weight plotted against SHP. Hewitt implies that the data refer to total plant weight. The data deals with single shaft configurations. The total weight of the three power plants for a triple screw 30,000 SHP vessel is found by entering Hewitt's curves at 10,000 SHP and multiplying the resultant by three.

Frankel and Simpson [31] provide graphical data for single screw plants. These data refer only to propulsion plant weight. The curve was read in the same manner as the Hewitt case.

Frankel and Marcus [31] present power plant weight information in equation form, for twin shaft total plant weight.

The NUS Corporation study [11] presents detailed weight group analysis for the Diesel Electric, Oil Fired Steam Turbo Electric, Gas Turbine Electric, and Nuclear-Steam Turbo-Electric (to be studied in Chapter 4). Figure 3.4 displays a summary of the results. The Total

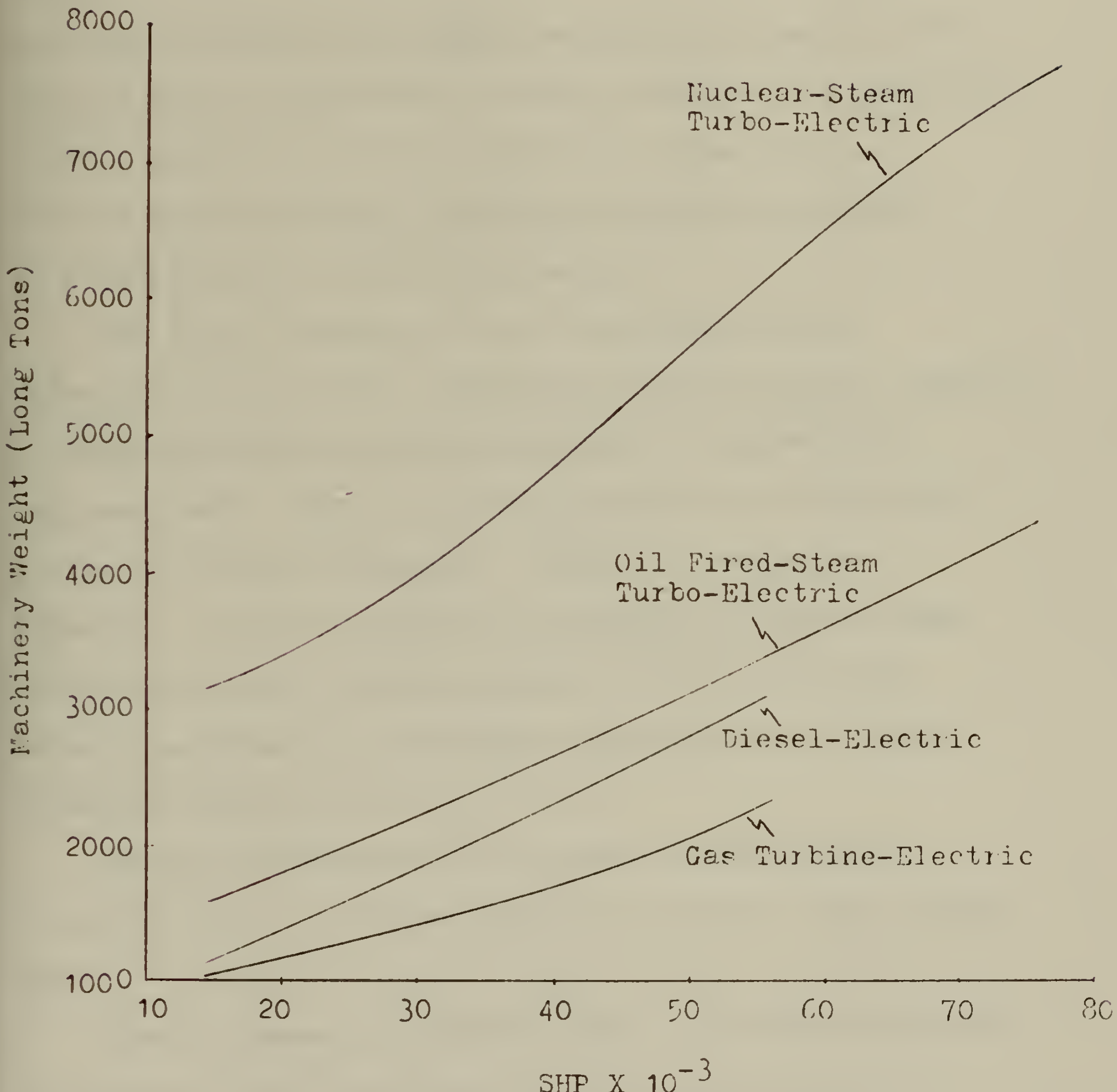
Figure 3.4

Total Comparative Machinery Weight

Vs.

S.H.P.

For U.S.C.G. Icebreaker Studies



Comparative Machinery Weight includes the propulsion plants' effect on all weight groups. Due to the thoroughness and specific application to icebreakers, the NUS weight data will form the basis for the synthesis model to be developed. Recent technology has allowed the switch from a DC-DC plant to a DC-AC plant. This provides a significant weight savings for Group 2. This savings will be accounted for when building the synthesis model. However, at this time only a comparison is desired and the Group 2 savings could be approximated as a constant for all of the plants.

The final parameter to be considered in the selection of the fossil propulsion plant is cost. Current literature contains several methods for estimating various costs. Some of these approaches are discussed in the following paragraphs. Table 3.3 contains a summary. A relative rating of machinery types by each author was again utilized because of the variations between methods (i.e. different year dollars, different overhead rates, etc.).

Marine Engineering [11] presents representative installed costs, plotting relative installed cost versus SHP.

Hewitt presents graphical data. He used normalized 1970 dollars and the curves include installation and acquisition costs, as well as a 66% overhead for installation. Hewitt implies total plant cost.

Frankel and Simpson provide graphical data for single screw ships as of 1967.

Frankel and Marcus present equations for total plant costs for twin screw vessels.

Femenia [29] also provides cost information in equation form. Femenia implies single screw vessels. The equations deal with 1971 dollars and produce total plant costs.

Table 3.3

RELATIVE COSTS OF PROPULSION PLANTS

Plants	References					
	M.E.	HEW.	F&S	F&M	FEM.	NUS
Diesel Electric						1
Diesel (Medium Speed)	1	1	1	3	1	
OFTE						3
Steam	2	2		2	2	
GTE						2
GT (Regenerative)	3	3		1	3	

Note--Plants are rated from 1 to 3 with 1 being the most economical.

NUS Corporation provides an in-depth cost analysis for icebreakers. The figures appear to be based on 1966 dollars.

At this point in the selection process, the impact of the operating profile fuel requirement was investigated,

using NUS specific fuel consumption rates (all-purpose) contained in Figures 3.1, 3.2, and 3.3. The results are displayed in Table 3.4. Diesel Electric is apparently the best choice based on weight, which will reflect in cost due to the size of the vessel. The Oil Fired Turbo-Electric and Gas Turbine Electric can be clearly eliminated from further analysis. Performing a sensitivity analysis and allowing a reduction of 10% in SFC causes the OFTE to drop to 23.3% and the GTE to 9.5%. The Diesel Electric is still a superior choice. If the relative costs are included, the diesel's superiority becomes even more readily apparent (there being only one of the six references who feel that diesel is not the most economical).

The differentiation between Diesel Electric and a CODOG-Electric is not as clear, primarily due to lack of data. The fuel calculation in Table 3.4 is based on operation of the more economical diesels at the 80% power level, only using the turbines at full power. This results in diesels sized to provide 24,000 SHP, and turbines capable of delivering 30,000 SHP. The plant weight has been determined by entering the Diesel Electric curve at 24,000 SHP, entering the Gas Turbine Electric curve at 30,000, and summing these weights. The validity of this procedure, particularly in light of the weight presented in Marine Engineering, is circumspect. Therefore, the 9% advantage of Diesel Electric in Table 3.4 is

Table 3.4

WEIGHT ESTIMATES BASED ON

OPERATING PROFILE

Profile	Plants							
	CODCG-E		DE		OFTE		GTE	
	SFC	TON	SFC	TON	SFC	TON	SFC	TON
10 Days @ 50%	.55	884	.55	884	.73	1173	.81	1302
45 Days @ 80%	.50	5786	.51	5901	.7	8100	.69	7984
7 Days @100%	.65	1463	.50	1126	.68	1530	.65	1463
Total Fuel		8133		7911		10,800		10,750
Plant Weight [11]		2400		1750		2200		900
Total		10,533		9661		13,000		11,650
% above DE		9.0				34.5		20.6

$$W_F = \text{SHP (SFC) DYS} \left(\frac{24\ \text{HR}}{\text{DY}}\right) \frac{1\ \text{Ton}}{2240\ \text{lbs.}}$$

Where:

WF = weight of fuel in tons

SFC = all purpose specific fuel consumption in

$$\left(\frac{\text{lbs.}}{\text{SHP-Hr.}}\right)$$

insufficient to rule out CODOG Electric. Although there is no cost data presented in Table 3.3, the added impact resulting from the turbines (initial capital costs, inlet and exhaust ducting, control systems, spare parts, training of personnel, etc.) will cause a distinct cost disadvantage. The Diesel Electric plant will be considered as the fossil propulsion plant for this thesis.

The author must point out that the combination of 45 days at 80 percent power and 7 days at 30,000 SHP, are the requirements which drive the solution toward diesels. A lower desired percentage of total power and/or a higher total power requirement would cause a shift in favor of the CODOG plant. The U.S.C.G. HAMILTION Class Cutter is an example.

3.3 Model formulation

Comparison of the weight estimation of plant and fuel in Table 3.4 (9661T) with the POLAR STAR plant and fuel in Table 3.4 (5987T), indicates that a larger hull form will be necessary. Therefore, a larger vessel must be developed. A simple math model, based on a weight balance and a geometrically similar (to the POLAR STAR) hull form, will be formulated. Utilizing a geometrically similar hull allows the assumption that stability criteria will be met, and that volume available will be equal to or greater than volume required. The only additional volume required will be due to the additional fuel.

Arrangement efficiency increases with the size of the ship, and fuel tanks can be located in "low priced real estate." Therefore, the volume available will probably exceed volume required.

The first step in the model formulation was to establish weight estimating procedures for each of the weight groups, as well as the loads. This was accomplished by combining data from the NUS study [11] (contained in Appendix D) and the POLAR STAR weight statement.

Table 3.5 consists of POLAR STAR characteristics, NUS and POLAR STAR weight groups, and model relationships. The POLAR STAR column has been taken from the weight statement. The margin was proportionally distributed to each weight group. The final column, entitled Model, was derived from the previous three columns.

Hull Structure (Weight Group 1) is based on the POLAR STAR value. The POLAR STAR has abandoned the practice, in the WIND Class and GLACIER, of using a truss structure to support transverse frames over six-tenths of the length. This has been replaced by a heavier grillage framing system. The grillage system was selected because it can experience local plastic deformation under overloads (such as ramming) and "not necessarily lose its ability to sustain additional loads at the design load level" [33]. The truss system, developing significant bending moments, when combined with axial loads, "could lead to a

Table 3.5

SHIP SYNTHESIS MODEL

POLAR STAR

LOA = 400 ft	LBP = .88LOA	C_p = .573
LBP = 352 ft	BMAX = LOA/4.79	C_m = .852
BMAX = 83.5 ft	BWL = LBP/4.51	C_B = .488
BWL = 78 ft		
T = 28 ft	T = BWL/2.786	C_w = .74
Δ_{FL} = 13,190 t	$\Delta_{FL} = 1.23\ \Delta_{WL}$	SAC = 31.05 = .542R
Δ_{28} = 10863 t		SA = 58.95 = 1.029R
		BA = 15 = .26179

D_{av} = 45 ft = .577 BWL

D_{MK10} = 44 ft

CN = 12.36

NUS Wt. Group	Machinery	35,000 SHP	POLAR STAR (With Margin)	MODEL (With Margin)
1	3400(partial)	5025	5000	404.5 T/CN
2	96 lb/SHP	107	60	.045 T/SHP
3	230	230	189	.537 T/Ft_{LBP}
4	100	85	25	25 T
5	900	1350	750	60.7 T/CN
6	680	750	633	51.2 T/CN
7	35	35	.83	1 T

LOADS - POLAR STAR

Fuel		
Diesel Oil	4379 T	(1,359,200 gal.($\frac{1T}{310.42\ gal}$))
JP5	150 T	(46,419 gal.)
Misc.	414 T	
POLAR STAR	13,190 T	

LOADS - Model

Fuel	10 days @ 50% SFC = .55	.02946 SHP (SFC) 24 (Days) $\frac{1}{2240}$
	45 days @ 80% SFC = .51	.1967
	7 days @100% SFC = .50	.0375
	15 days drifting	
	77 days = 48 full power	.2637 SHP
	95% Tail Pipe Factor	.2776 SHP

Misc. JP5 + Misc. = 150 + 414 = 564 T

progressive failure of adjacent frames through elastic instability, and subsequent collapse of the shell framing" [33].

Propulsion Machinery (Weight Group 2) estimating relationship is based on the NUS study. The lower figure for the POLAR STAR is a result of using lightweight gas turbines to achieve 60,000 SHP. The all-diesel NUS case has more applicability in this study.

The model estimating relationship for the Electric Plant (Weight Group 3) results from the POLAR STAR data and the NUS concept of relating weight to length overall. The POLAR STAR weight is considered valid.

Communication and Control (Weight Group 4) is fixed as a constant. This was the technique used in the NUS study. The NUS study selected "rather arbitrary values" and included a Mark 56 Gun Fire Control System. The POLAR STAR contains no gun control system and utilizes the most modern microcircuitry, resulting in a weight reduction.

The Auxiliary Systems (Weight Group 5) estimator is taken from the POLAR STAR data and related to the cubic number as in the NUS study. Auxiliaries (boilers, evaporators, air conditioning, etc.) are also a function of the crew size. The NUS values relate to a 400 foot ship with a complement of 450 men. The POLAR STAR (352 feet) has a total complement of 164 men, yielding a smaller Weight Group 5. This complement level is felt to be consistent with current U.S.C.G. manning philosophy.

Outfit and Furnishings (Weight Group 6) estimating relationship is again derived from POLAR STAR data, utilizing the cubic number as the critical ship's parameter. Outfit and Furnishings is also a function of complement. The argument presented for Weight Group 5 also applies to this weight group.

Armament (Weight Group 7) is taken as a constant. The POLAR STAR value of approximately one ton is consistent with the original non-military mission statement. No change in this philosophy by the U.S.C.G. is foreseen in the near future.

To complete the weight portion of the model, the loads must be considered. Table 3.5 presents the POLAR STAR loads as well as the model loads. The JP5 and Miscellaneous loads were held constant. The fuel loading was adjusted to reflect the additional endurance requirements.

The relationships for length, beam, draft, displacement, depth, and shape are contained in Table 3.5. In order to maintain geometrical similarity, these ratios are based on the POLAR STAR.

The final element necessary to complete the model is a method of dealing with shaft horsepower. The POLAR STAR open water speed power curve is presented in Figure 2.9. The similarity in hull form should result in a similar curve for the model. Therefore, an equation was developed to describe the curve. The equation for a Destroyer with $.7 < V/\sqrt{L} < 1.2$ was used as a base [34]:

$SHP = .00342\ V^{3.33}\ \Delta^{.6117}$

The following equation was developed by iteration, the errors are also noted:

$SHP = .0025\ V^{3.54}\ \Delta^{.62}$

V = 13 kts	SHP_M = 9,516	SHP_{PS} = 9,000	+5.7%
V = 17 kts	SHP_M = 18,031	SHP_{PS} = 18,000	+.17%
V = 21 kts	SHP_M = 47,936	SHP_{PS} = 60,000	-20.0%

M - model

PS = POLAR STAR

The three values are represented as squares in Figure 2.9. The points fit the shape of the curve fairly well, particularly at 17 knots, the required open water speed. The equation was employed to estimate open water shaft horsepower for the model, 24,683 SHP.

The shaft horsepower required to break six feet of ice in the continuous mode was assumed and later validated using Milano and Vance in the same manner as Table 2.2 and 2.3. The assumed values were based on 30,000 SHP, found in Section 2.3. Section 2.3 also demonstrated that the icebreaking, not the open water, requirement drives the selection of shaft horsepower. Therefore, the open water equation serves only to produce a lower limit for the installed shaft horsepower.

The completed model is, at this point, capable of generating a ship system that satisfies a weight balance.

The inputs for the model are length overall and installed shaft horsepower. Table 3.6 presents iterations leading to an engineering approximation for the smallest vessel (henceforth known as M1) capable of meeting a weight balance. To constitute a feasible solution, the ship must pass the weight balance established by the model and be capable of breaking six feet of ice in the continuous mode. The next phase of the problem is to investigate the ship's icebreaking capability.

3.4 Ship validation

Milano and Vance are employed to assist in the validation of the vessel's ability to meet the icebreaking capability. Use of Milano requires inputs (Table C.2) from the ship that were not developed by the weight balance model. The first of these is the propeller diameter, PDIA. Two equations are available, which are in general agreement:

$$PDIA = 8.2\ (P/1000Z)^{2/7} \pm 20\% \qquad [5]$$

where

P = SHP

Z = Number of propellers

$$PDIA = MIN.\ [H - 9;\ \frac{.625B - 4}{3}] \qquad [35]$$

where

H = Draft

B = Max beam @ DWL

Table 3.6

SHIP SELECTION

LOA	400	450	475	500
LBP	352	396	419	440
BMAX	83.5	93.9	99.4	104.4
BWL	78	87.8	92.9	97.6
T	28	31.5	33.3	35.0
ΔWL	10,719	15,270	18,073	20,976
ΔFL	13,184	18,783	22,230	25,800
SHP	30,000	31,000	33,000	35,000
DAV	45	50.7	53.6	56.3
CN	12.36	17.6	20.9	24.2
W1	5000	7119	8454	9789
W2	1290	1333	1419	1505
W3	189	213	225	236
W4	25	25	25	25
W5	750	1068	1269	1469
W6	633	901	1070	1239
W7	1	1	1	1
Light Ship	7888	10,660	12,463	14,264
Loads				
Misc.	565	565	565	565
Fuel	7911	8175	8702	9230
TOTAL	16,364	19,400	21,730	24,059
ERROR	-19.4%	-3.2%	+2.3%	+7.2%
			M1	

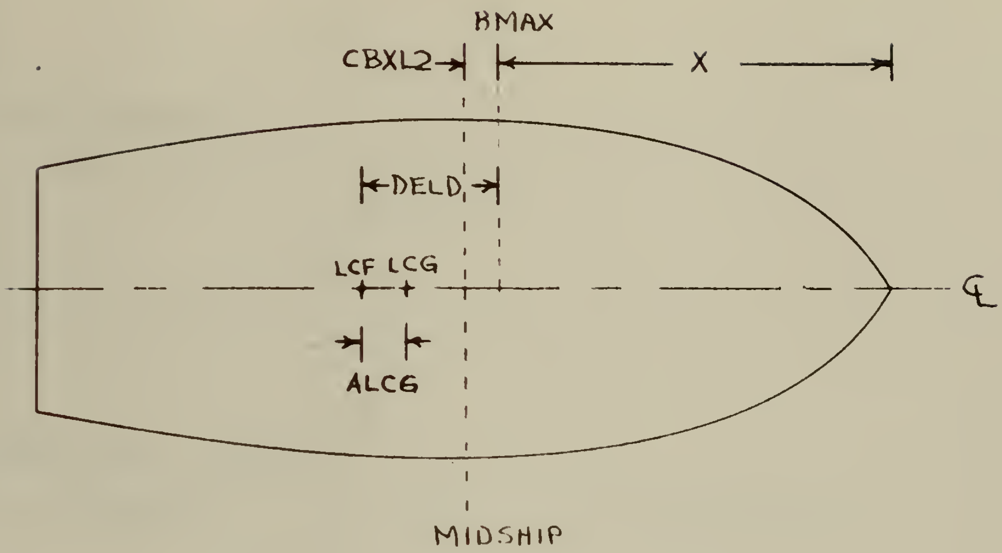
The remaining input parameters were obtained, relying on geometrical and, consequently, water plane similarity, by parameter to length ratios. Figure 3.5 illustrates X, DELD, CBXL2, and ALCG. Geometric similarity implies no change in CX, CW, ALPHA, CBI, CXI, GBETA, and GDELT. The necessary values are listed in Figure 3.5. The unchanged parameters can be found in Appendix D, POLAR STAR Input Data Base.

The needed inputs for Milano's program are now available. The program was executed, the results are contained in Table 3.7. This is the same technique for finding SHP as presented in Table 2.2. Again, the most severe set of ice conditions is selected. In this case, the resistance is found by extrapolation of the base run using the differential percentage variances noted in Table 2.2. The resistance is then checked with Vance as Table 2.3. The M1 required 32,197 SHP to meet the icebreaking constraint. The installed 33,000 SHP exceeds the required 32,197 SHP, even if an allowance for snow cover is included. The thrust to weight ratio for M1 is 1.89×10^{-2}. Investigation of Table 2.5 indicates that the ratio is acceptable but definitely approaching the lower limit. No further reduction is SHP is deemed prudent.

The goals of this chapter have been met. The minimum impact conventional propulsion system was identified as Diesel Electric. Subsequently, the resulting ship system

Figure 3.5

Geometrical Scaling



POLAR STAR	RELATIONSHIPS	M1 (Feet)
X = 140.8	X = .4LBP	167.6
DELD = 33.3	DELD = .094LBP	39.6
CBXL2 = 35.2	CBXL2 = .1LBP	41.9
GML = 350.1	GML = .9946LBP	416.7
ALCG = 1.35	ALCG = .0038LBP	1.61

Table 3.7

M1 SHAFT HORSEPOWER VALIDATION

MILANO

Fixed Parameters

σ_c = 50,000 psf.

S.G. = .92

V = 6 ft./sec.

h = 6 ft.

Computer Run - Base

σ_f = 15.000 psf $\qquad$ μ = .2 $\qquad$ R_1 = 624,373 lbs.

SHP Calculation

Select the most stringent case σ_f = 20,000 μ = .3

Use increases from base noted in Table 2.2

1. 51% for 5000 psf increase in σ_f

2. 20.4% for μ= .3 in lieu of .2

R = 1.51 (1.204) R_1 = 1,135,135 lbs.

Service allowance (SA) = 1.3

PC = 0.5

$$SHP = SA \frac{EHP}{550(PC)} = 1.3 \frac{RV}{550(PC)} = 1.3 \frac{(1,135,135)}{550(.5)}$$

SHP = 32,197

Table 3.7 (cont.)

VANCE

$$R_{(ice)} = C_s \rho_\Delta g B h^2 + C_B \sigma_f B h + C_V \rho_i V^2 L h^{.65} B^{.35}$$

$$= 16.91(.1552)32.2(92.9)(6)^2 + .034(15,000)92.9(6)$$

$$+ .165(1.7847)(6)^2\ 419(6)^{.65}(92.9)^{.35}$$

$$= 282,625 + 284,274 + 69,530$$

$$= 636,428 \text{ lbs. vs. } 624,373 \text{ lbs. MILANO}$$

THRUST

$V = 0$ ft/sec $h = 1$ ft

Thrust= 764,339.7 lbs

Weight= 18,073(2240) = 40,483,520 lbs.

Thrust/Weight = 1.89×10^{-2}

was formulated and found capable of meeting the icebreaking requirement.

CHAPTER IV

NUCLEAR PLANT SELECTION

The optimum fossil fuel propulsion plant and corresponding ship system were formulated in the preceding chapter. This chapter is devoted to identifying a nuclear propulsion plant with proven marine capabilities and to formulate a compatible ship system. The resulting vessel shall satisfy the previously noted mission requirements, as well as the Plant Requirements noted in Section 3.1. The task was approached in the following manner: investigate various nuclear alternatives, perform necessary modifications to math synthesis model, and validate model output.

4.1 Reactor selection

A marine nuclear power plant is composed of several basic components: nuclear steam generator (including reactor), main turbines and condensers, and turbine generator sets. The output of the turbines in this study will drive the equivalent electric system to the Diesel Electric plant discussed in Chapter 3. This section addresses the power source and treats the electric power transmission system as a constant.

The major component or variable in the nuclear power plant is the reactor type. The pressurized light water thermal reactor (PWR) utilizes water under pressure as coolant and moderator, high power density, automatic load following, and is fueled with low U_{235} enriched uranium oxide. The PWR is installed in all presently operating marine applications [27]. It has a history of reliability and safety with shore side central power station installations and Navy submarines. The PWR has been selected as the nuclear power source for the Nuclear-Steam Turbo-Electric propulsion plant. To facilitate weight and cost estimation, and for completeness, the PWR is described further in the following paragraphs.

The light water pressurized reactor consists of eight essential subsystems [36]: the core, the moderator and coolant system, the pressure vessel, the heat exchanger system, the control rod drives, the containment and shielding, instrumentation, and the various auxiliary systems. Comparing a Nuclear-Steam Turbo-Electric with an Oil Fired Steam Turbo-Electric, the nuclear reactor is analogous to the boiler. However, the nuclear produced steam is of poorer quality, being either saturated or only slightly superheated. This steam condition necessitates larger, slower, more expensive main turbines than a comparable oil fired steam system. In addition, maintenance and operating costs are aggravated as a result of increased blade and nozzle erosion.

The eight subsystems comprising the reactor system are called the "nuclear" components by Caroussis [36]. The "non-nuclear" components include: main turbines and condensers, turbine generator sets, take-home engine, emergency diesel generator set, the propulsion generators and main motors, etc.

In addition to the items noted previously, there are some components, which must be considered, that are directly related to the nuclear plant and not contained in Weight Group 2. Some of these "miscellaneous" items are [36]:

Hull changes--collision barrier, strengthened and deeper double bottom floors below the reactor, reinforced longitudinals, etc.

Health physics and water chemistry laboratories

Hot machine and instrumentation shops

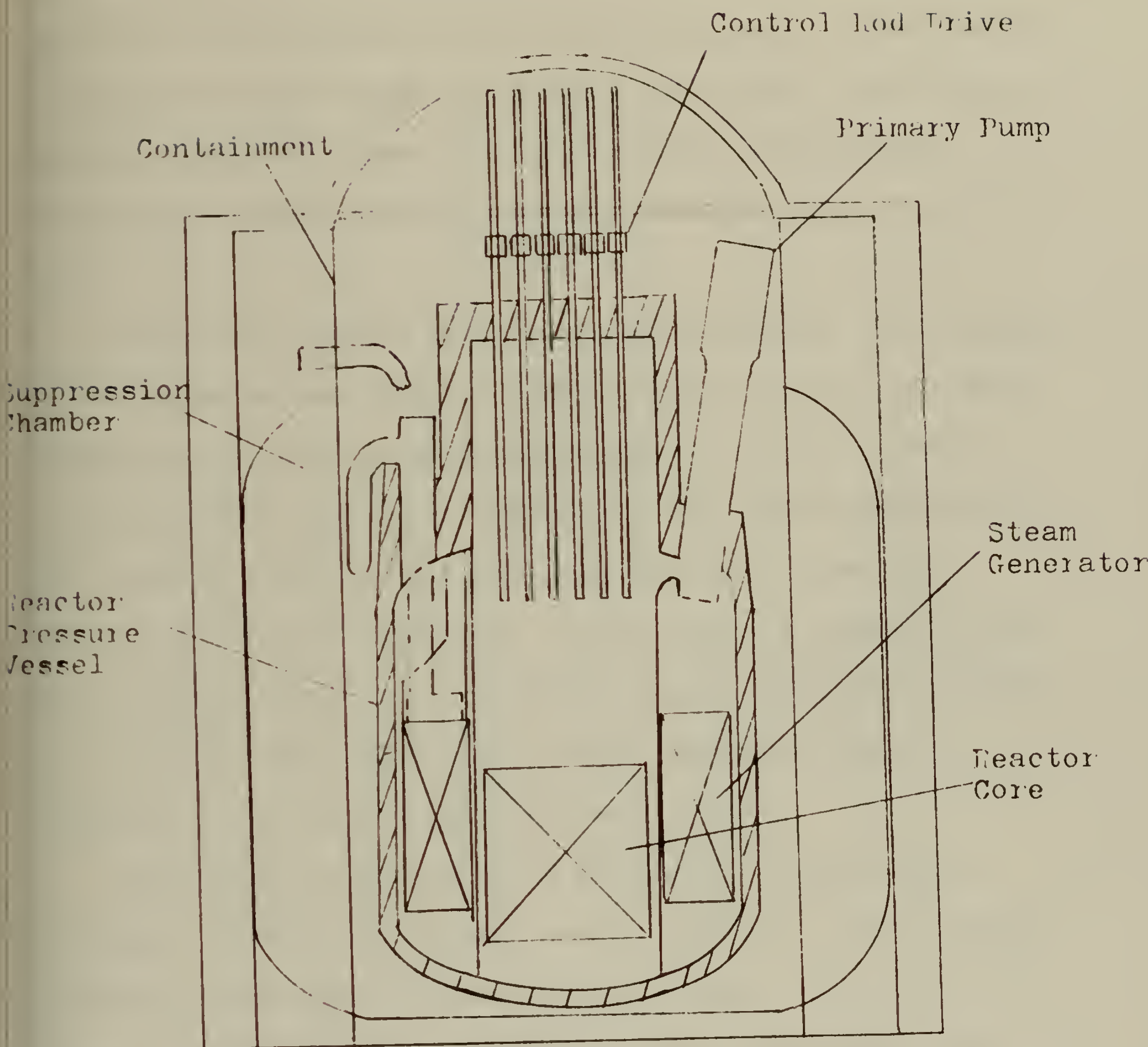
Decontamination facilities

Caroussis [36] mentions these points as they pertain to tankers. Hull changes will have a minimum impact on an ice strengthened icebreaker. This concludes the general discussion of PWR plants.

At this point, a specific nuclear plant, the Babcock and Wilcox CNSG was chosen as being representative of PWR plants. The Consolidated Nuclear Steam Generator (CNSG) is the result of a jointly sponsored MARAD-Babcock and Wilcox program [36] (Figure 4.1). Caroussis [36]

Figure 4.1

CNSG Pressurized-Water Reactor



Cross Section

claims that the CNSG combines both low weight and cost including good reliability experience, as well as the expertise and experience of the manufacturer. The German nuclear ship OTTO HAHN is powered by a CNSG. The Global Marine study [37] used a CNSG reactor. The CNSG is described in more detail in the following section [27, 36, 37].

The Consolidated Nuclear Steam Generator is a compact pressurized water reactor, containing the core and steam generator inside the reactor vessel. An electrically heated pressurizer is connected to the vessel externally. The reactor fuel is low enriched uranium oxide (UO_2) pellets clad with Zircaloy. Reactivity is controlled by utilizing control rods and fixed lumped burnable poisons. The control rod drives and primary coolant pumps are located on the top of the pressure vessel.

The steam generator is a helically coiled, once through, forced circulation unit, located in the annulus between the reactor vessel wall and core. The coils of tubes are connected to feedwater and steam tube sheets in four separate circuits. Each section may be operated independently if isolation is necessary, increasing reliability. The once-through design using counterflow heat transfer with tubeside boiling, allows the system to produce superheated steam at a constant pressure. Superheated steam provides higher turbine efficiencies, fewer

maintenance problems, and a greater margin for load changes without moisture carry-over to the turbines [27, 37].

The CNSG utilizes lead, water, concrete, and steel in the shielding design. The high mass materials, lead and steel, act to absorb gamma radiation and to slow fast neutrons by inelastic collisions. The water and concrete constitute materials with high capture probability for the slowed neutrons.

The next section of this chapter will look at the modifications of the math model to accommodate the nuclear system.

4.2 Model modification

The simple math synthesis model developed in Chapter 3 was formulated on the basis of Diesel Electric data. It should be obvious that revisions are necessary to render the model appropriate for nuclear applications. Chapter 3 utilized overall weight groups to develop the model. The NUS study [11] conducted in-depth machinery weight group analysis for different propulsion plants. Table 4.1 contains a summary of Weight Group impacts by Diesel Electric and Nuclear-Steam Turbo-Electric [11] propulsion plants for different vessel sizes. This table also includes the necessary parameters to form model relationships (i.e. LBP, CN, and SHP). The initial model modifications are for a ship with: CN = 12.36, SHP = 30,000, and

Table 4.1

MACHINERY WEIGHT FOR DIESEL

AND NUCLEAR PLANTS

LBP		300	400	500	600
		9.3	16.8	24.7	31.6
SHP		15,000	35,000	55,000	75,000
W_1	DE	70	128	183	
	NSTE	600	925	1264	1600
W_2	DE	850	1580	2389	
	NSTE	2200	2857	4186	5100
W_3	DE	160	231	296	
	NSTE	200	284	378	470
W_5	DE	100	146	226	
	NSTE	90	135	198	270

DE - Diesel Electric

NSTE - Nuclear-Steam Turbo-Electric

Ft_{LBP} = 352, the POLAR STAR hull parameters. The appropriate Diesel Electric model relationships are reproduced from Table 3.5 for the reader's convenience.

W_1 404.5T/CN

W_2 .045T/SHP

W_3 .537T/Ft_{LBP}

W_5 60.7T/CN

4.2.1 Light ship requirements

The model relationship for W_1 is based on the POLAR STAR. Figure 4.2 displays the Diesel Electric (left vertical axis) and Nuclear Steam Turbo Electric (right vertical axis) propulsion plant impacts on Weight Group 1 based on the NUS study [11]. The following engineering assumption is made: the ratio of propulsion plant influence in W_1 to total W_1 is equal for the NUS ship and the POLAR STAR for each type of plant. This assumption facilitates the approximation of the modified POLAR STAR (M1) propulsion plant impact on W_1. The nuclear model (M2) relationship for W_1 is then determined by: subtracting the proportional M1 diesel electric plant impact and adding the NUS nuclear plant (N_1) to the M1 Weight Group 1 relationship.

$$W_1[M2] = W_1[M1] - \frac{DE_1[NUS]}{W_1[NUS]} (W_1[M1]) + N_1[NUS]$$

$$W_1[M2] = 404.5T/CN - \frac{7.28T/CN}{300T/CN} (404.5T/CN)$$

$$+ 59T/CN = 453.7T/CN$$

Figure 4.2

Propulsion Impact on W_1

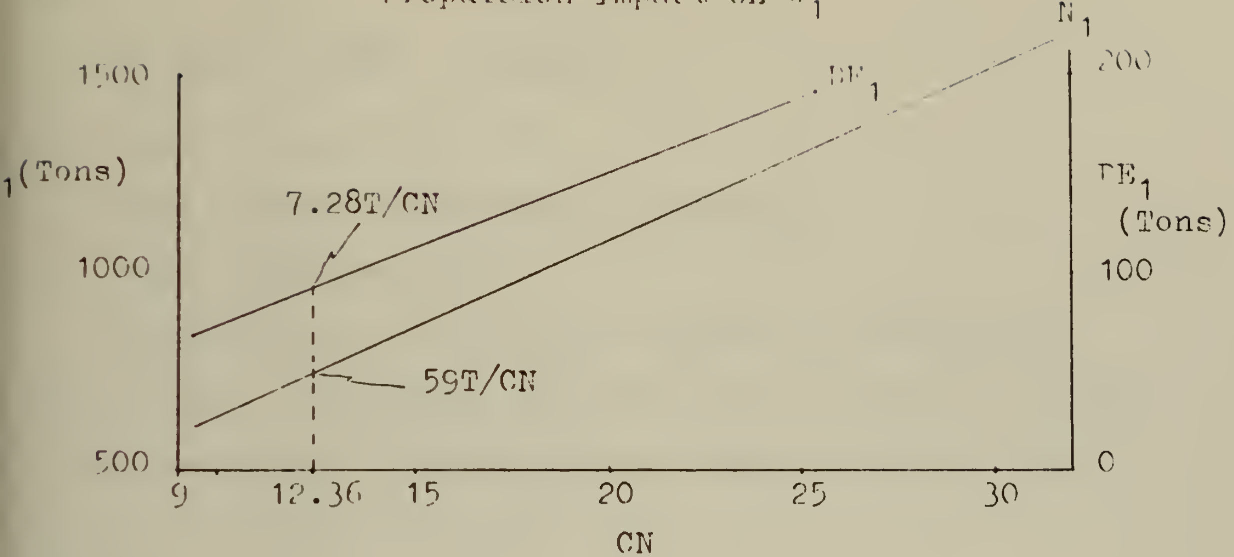
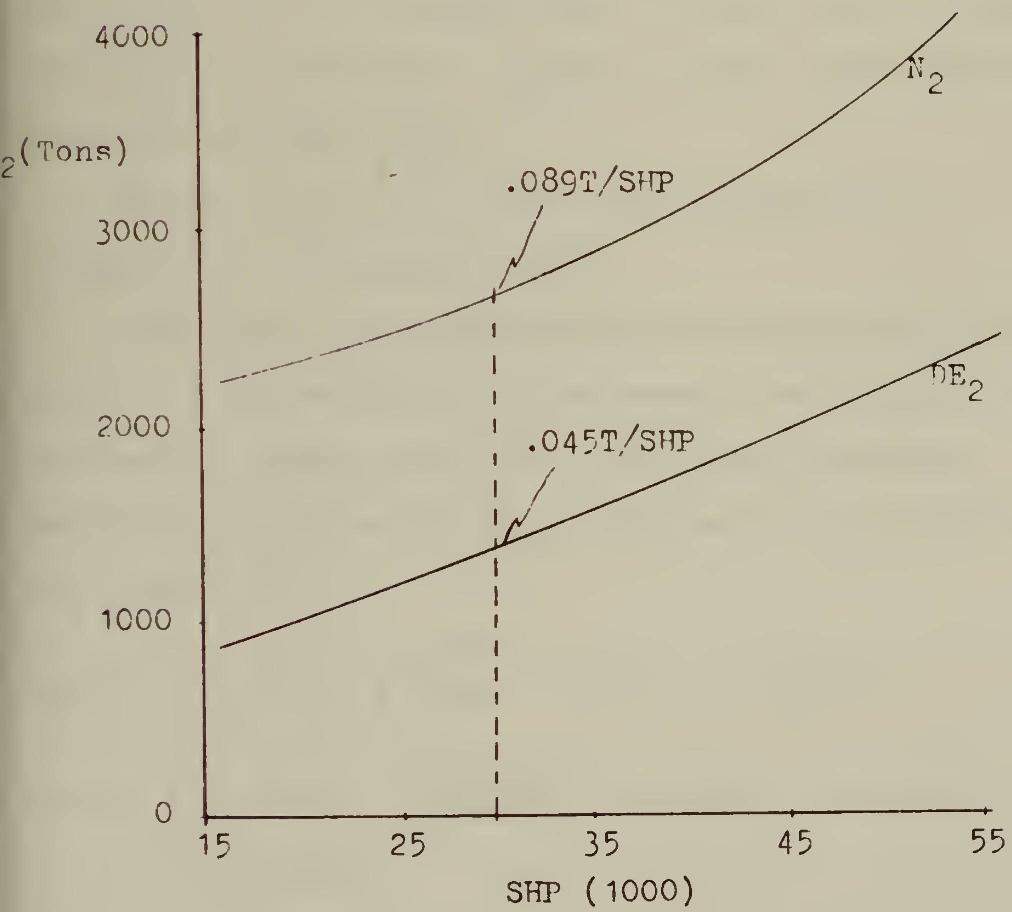


Figure 4.3

W_2 vs. SHP



where:

$$W_1[NUS] = \frac{5025T}{16.8} = 300T/CN$$

[NUS] - value from [11]

[M1] - value from Chapter 3 (fossil)

[M2] - nuclear value

Weight Group 2 is entirely composed of propulsion plant impact. Therefore, W_2[NUS] can be replaced by W_2[M2] in the new model. W_2[M2] is then easily determined from Figure 4.3 to be:

.089T/SHP

The NUS study computed W_3 separately for the diesel plant and for the nuclear plant. W_3[M1] can be replaced by W_3[M2] in the model. W_3[M2] is then found from Figure 4.4 to be: .688T/Ft_{LBP}.

Weight Groups 4, 6, and 7 are assumed to be independent of propulsion plant.

The final light ship modification deals with Weight Group 5. The model (M1) relationship for Weight Group 5 is based on POLAR STAR data. The proportionality assumption utilized for W_1 is applied to W_3, with the aid of Figure 4.5.

$$W_5[M2] = W_5[M1] - \frac{DE_5[NUS]}{W_1[NUS]}(W_5[M1]) + N_5[NUS]$$

$$W_5[M2] = 60.7T/CN - \frac{8.69T/CN}{80.4T/CN}(60.7T/CN) + 8.82T/CN$$

$$= 62.96T/CN$$

Figure 4.4

W_3 vs. FT_{LBP}

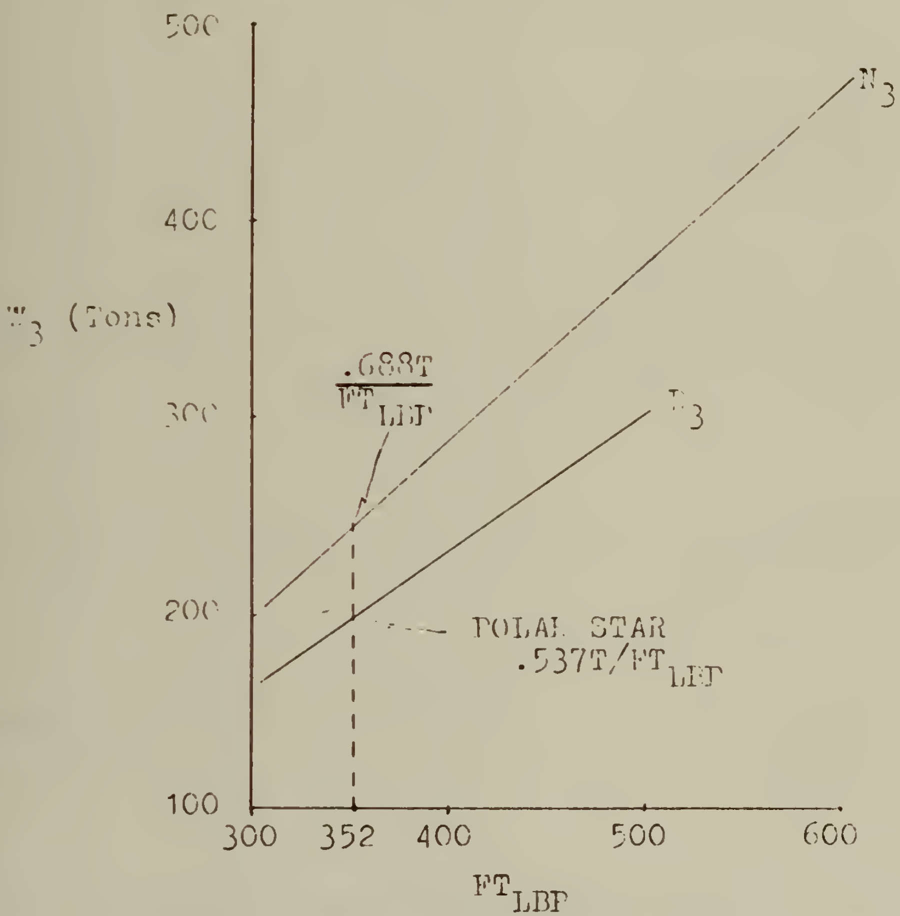
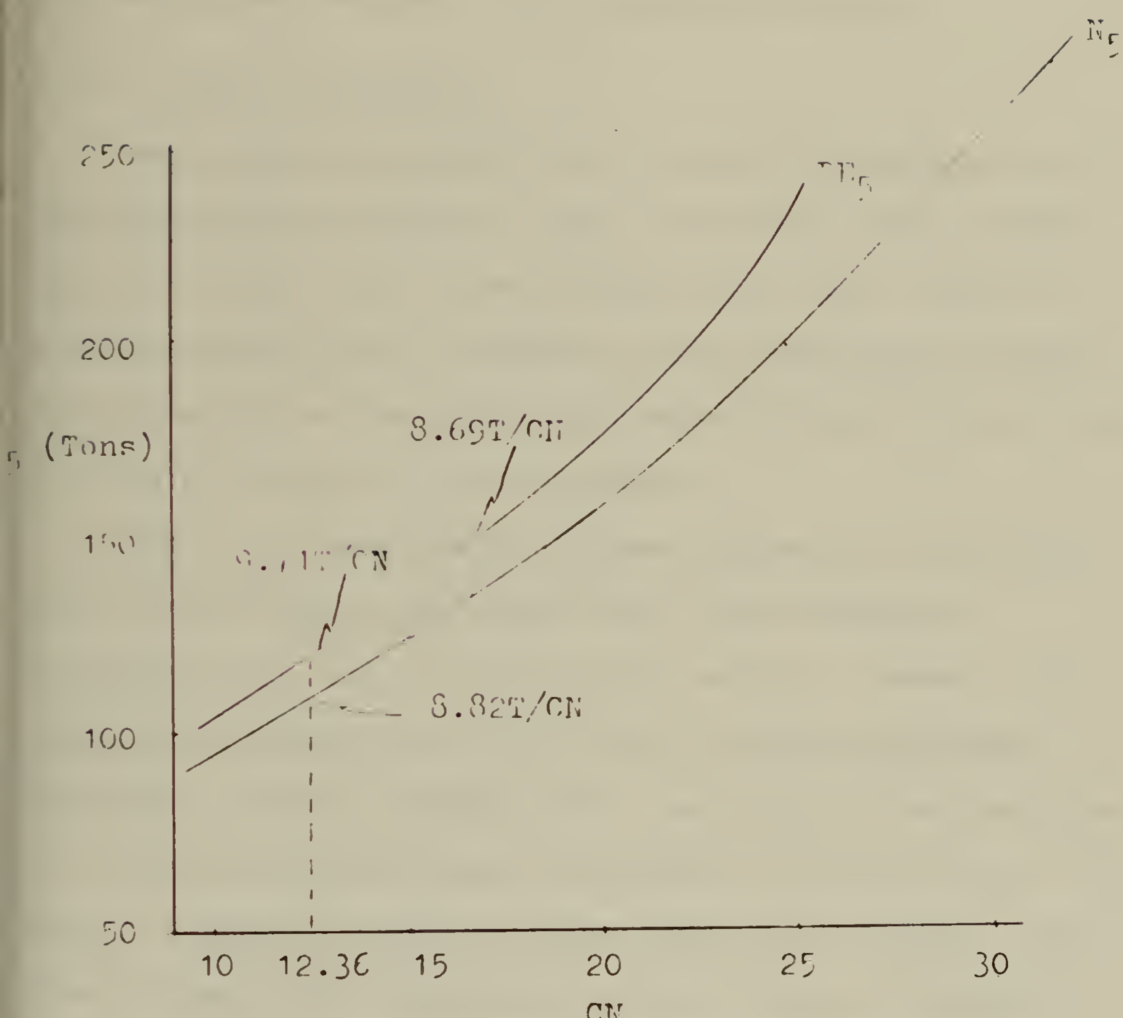


Figure 4.5

Propulsion Impact on W_5



where:

$$W_5[NUS] = \frac{1350T}{16.8} = 80.4T/CN$$

A general validation of the nuclear propulsion plant weight was performed. Table 4.2 indicates reasonable results and sufficient accuracy for conceptual design.

4.2.2 Load requirements

The load requirements for a nuclear icebreaker will be considerably different than the conventional sister ship. In this study , the JP5 fuel (150 Tons) and the Miscellaneous Loads (414 Tons) were considered constant. Determination of the fuel requirements entail identification and basic sizing of fuel consumers.

The U.S. Coast Guard, in the NUS study, felt that one nuclear plant with sufficient loop redundancy eliminated the need for a second reactor. However, an emergency propulsion or "take home" system was deemed necessary. More recently [43], the U.S.C.G. has required all nuclear powered ships, regardless of type or size, to have a complete alternate power propulsion system. This thesis makes the assumption that the U.S.C.G. has not altered its views with regard to multiple reactors or "take home" systems. An emergency diesel generator is standard practice. The author feels that an auxiliary source of steam for propulsion, hotel, and heating requirements would indeed prove most prudent. This is

Table 4.2

NUCLEAR PLANT WEIGHTS

Reference	Plant (W_2)	Total Impact (W_1, W_2, W_3, W_5)
Model	2670 T	3750 T
Caroussis [36]	2265 T	2874 T
Harrington [27]	1313 T	----
Engineers Digest [39]	----	4069 (35,000 SHP)
LENIN [25]	2980	---- (44,000 SHP)
Oakley [25]	3410	---- (35,000 SHP)

Vessel for Model, Caroussis, and Harrington

LBP = 352 feet

CN = 12.36

SHP = 30,000

in light of the severe operational requirements and conditions placed upon an icebreaker.

4.2.2.1 Minimum steam requirements

Sizing the auxiliary boiler requires determination of the propulsion, hotel, and heating requirements. The NUS study [11] computes the hotel steam requirements (lb/hr) based on the complement and claims agreement with existing icebreakers. The data takes the following form (Figure 4.6):

Q_H(lb/hr) = 4X + 810 X = Complement

The heating steam requirements (lb/hr) can be determined by use of the U.S. Navy's [11] equation:

Q_S(lb/hr) = ABC/K

where:

A = LBP

B = Breadth

C = Depth to Main or Weather Deck

K = 165.6 for GLACIER (use for this study as an approximation)

The mission requirements provide no guidance with regard to emergency propulsion systems. The NUS study [11] proposed an auxiliary boiler capable of driving the ship at 9 knots with the emergency diesel generator supplying the electrical load. Therefore, the author will assume that the U.S.C.G. has not altered its views. Obviously, the

Figure 4.6

Q_H Vs. Complement

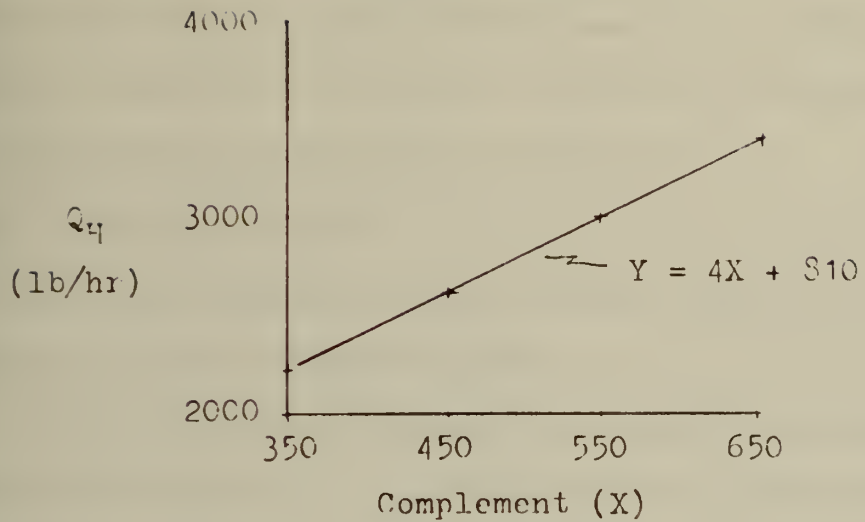
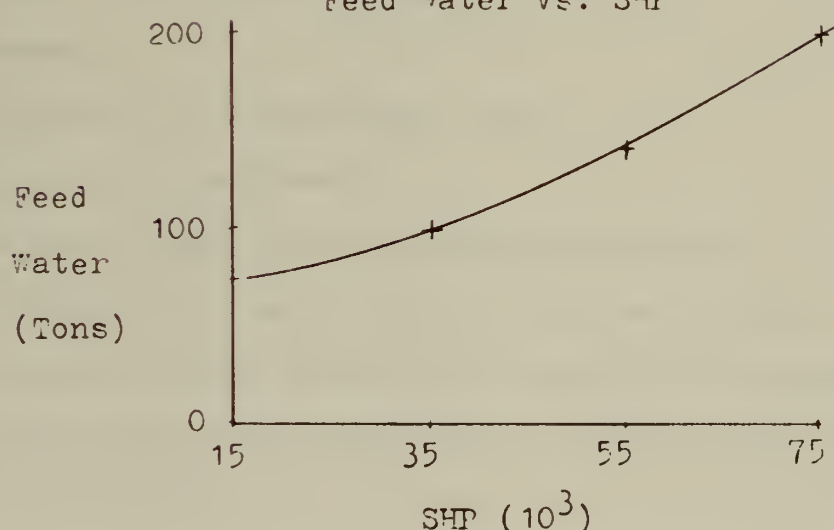


Figure 4.7

Feed Water Vs. SHP



boiler could be utilized to drive a turbine generator and/or evaporator with an expected degradation in ship speed. The SHP required to achieve 9 knots can be approximated by the equation developed in Chapter 3:

$$SHP = .0025\ V^{3.54}\ \Lambda^{.62}$$

If the POLAR STAR hull form is used, Figure 2.9 can provide a more accurate estimate of SHP.

The auxiliary boiler must be sized to include the three components, Q_S, Q_H, and propulsion equivalent to 9 knots.

4.2.2.2 Non-propulsive electrical requirement

An empirical relationship presented in SNAME Technical and Research Bulletin No. 3-11 [40] provides an estimation for 24-hour average total electric load:

$$KWavg\ 24 = A(SHP) + BN + 6.3\ \sqrt{N}$$

where:

A(SHP) is the propulsive load

N = total complement

B = 1.0 for fully air conditioned ship

B = 0.5 for a ship with no air conditioning

The equation can be simplified to provide non-propulsive electrical load. Assuming full air conditioning:

$$KW = N + 6.3\ \sqrt{N}$$

A conservative approach would be to provide the emergency generator with a capacity determined by the non-propulsive load.

The relationship for determining the minimum fuel requirements are summarized in Table 4.3. These provide the fuel oil load necessary for the modified model.

4.2.2.3 Reserve feed water requirements

The final load relationship to be developed deals with reserve feed water. Figure 4.7 displays a plot of data from [11]. These data are nuclear feed water requirements, not oil-fired steam requirements.

This concludes the nuclear model modification phase of Chapter 4. The next phase of ship system solution is to utilize the nuclear model to determine the smallest reasonable vessel capable of meeting the mission requirements.

4.3 Ship determination and validation

This section will identify a solution to the weight balance math model that satisfies the mission requirements. In the nuclear system the endurance requirement does not necessitate the large fuel capability seen in Table 3.6. With this perspective in mind, the POLAR STAR hull was utilized in the first nuclear model run (Table 4.4). The complement was increased by six men as a conservative estimate of increased nuclear manning requirements. Model result M2 indicates that the required weight is less than the full load waterline (2.5%) of the POLAR STAR hull form, including an endurance of 45 days (9720 miles) on

Table 4.3

DIESEL FUEL OIL REQUIREMENTS

Steam

Assumptions

1. Feedwater at $280^{o}F$ yields $h_1 = 249$ BTU/lb [41]
2. Steam at 685 psi yields $h_2 = 1383$ BTU/lb [41]
3. SFC = .5lb/SHP-HR
4. 1KW-HR = 3413 BTU [42]
5. 1HP = 1.341 KW [42]

$Q = Q_H + Q_S + 10\%$

$Q = 4N + 810 + LBP(B)\ Dav/165.6$

$Power_1\ (HP) = Q\ \Delta h = lb/HR(BTU/lb)(1KW\text{-}HR/341BTU)*$

$(HP/1.341\ KW)$

$= Q\ (1383-249)/[3413(1.341)] = .248Q$

$Power_2\ (SHP) = .0025\ V^{3.54}\ \Delta^{.62}$

$Power_T = (Power_1 + Power_2)\ 1.10$

$W_F(Tons) = SFC(Power_T)\ (Days)(24/2240)$

Electrical

Assumptions

1. SFC = .4

$KW = N+6.3\sqrt{N}$

$HP = 1.341\ KW$

$W_F(Tons) = SFC\ (HP)\ Days\ (24/2240)$

$= .4(1.341KW)\ Days\ (24/2240)$

$= .0057(KW)\ Days$

$= .0057(N+6.3\sqrt{N})\ (Days)$

Table 4.3 (cont.)

Diesel Oil Load

Load = Steam + Electric

Let X = 170, LBP = 352, B = 78, Dav = 45, 9Kts. = 5600 SHP

Days	30	45	60
Electric Load (T)	43.12	64.67	86.23
Steam Load (T)	1382.43	2073.64	2764.86
Total	1425.6	2138.3	2851.1
95% Tail Pipe	1500.6	2250.9	3001.1

Table 4.4

SHIP SELECTION

LOA	400
LBP	352
BMAX	83.5
BWL	78
T	28
ΔWL	10,719
ΔFL	13,184
SHP	30,000
DAV	45
CN	12.36
Complement	170
W_1 (453.7T/CN)	5607.7
W_2 (.089T/SHP)	2670.
W_3 (.688T/FT_{LBP})	242.2
W_4	25
W_5 (62.96T/CN)	778.2
W_6 (51.2T/CN)	632.8
W_7	1
LIGHT SHIP	9,956.9
LOADS	
Fuel Oil	2,250.9
Reserve Feed Water	90
Misc.	564
TOTAL	12,862
ERROR	322T 2.5%

M2

emergency boiler and generator. When considering this result, the reader must be reminded of several important points:

--The math model performs only a weight balance.

--The model does not address total ship volume, nor propulsion plant volume.

--The linearities presented by [11], used in developing the nuclear model, require further validation.

The M2 result is accepted as the nuclear ship system solution. The hull form and powering have been shown (Section 2.3) to meet the six-foot continuous mode icebreaking requirements, to be capable of attaining 17 knots in open water, and to have no serious degradation in the ramming mode. The model indicates that the nuclear system will yield a displacement 2.5% less than full load displacement (12862 vs. 13184 tons).

The objectives of this chapter have been accomplished.

The Babcock and Wilcox Consolidated Nuclear Steam Generator was selected as the nuclear propulsion plant. The weight balance math model was modified to accommodate the nuclear system. Finally, a ship system was selected. This resultant vessel will be compared with the conventional output of Chapter 3 in the following chapter.

CHAPTER V

ECONOMIC ANALYSIS

An economic comparison of the fossil (Chapter III) and nuclear (Chapter IV) powered ship systems will be presented in this chapter. The identification of an optimum economic choice is the goal of Chapter V. Several phases were involved in the attainment of this goal: investigation of conventional and nuclear cost estimating techniques for initial investment and operating costs, comparison of the conventional and nuclear ship systems based on a discounted cash flow, and brief discussion of nuclear impacts that are not direct costs.

5.1 Cost estimation for conventional ships

5.1.1 Capital cost

The first method to be discussed was developed by Yuhas, a U.S.C.G. officer, in 1967 [44]. This method was developed primarily to provide estimates for acquisition and construction of fossil fueled polar icebreakers, but to have general applicability to U.S.C.G. shipbuilding programs. Yuhas does not deal with precontract design work nor operating costs of the resultant ship. The Yuhas-developed Ship Costing Procedures were utilized by the U.S.C.G. to estimate the costs involved in acquisition and construction of the POLAR STAR Class icebreakers.

Yuhas apparently based the estimating relationships on 1967 dollars. To facilitate cost comparison between several estimating procedures, a common year and inflation rate must be selected. The inflation rate will be used to bring all of the methods to a common base year. All costs will be inflated to 1976 dollars, the base year. The base year will serve as the zeroth year for the discounted cash flow analysis. The inflation rate is assumed equal for labor and materials. The average hourly wage quoted by Caroussis [36], in late 1974, was \$6.15. Yuhas quotes \$3.50 in early 1967. The simple compounded inflation rate is therefore determined to be [46]:

Amount at Compound Interest = $P(1 + i)^n$

where:

P = original principal

i = interest rate (inflation)

n = number of periods (years)

= approximately 8 years

P = \$3.50/hour

Amount at Compound Interest = \$6.15/hour

$$i = \left[\frac{6.15}{3.50}\right]^{1/8} - 1$$

$i = 1.073 - 1 = 7.3\%$

Obviously, this represents a simplistic approach to dealing with inflation. Labor and material do not inflate at the

same rate. Table D.4 in Appendix D presents inflation indicators for labor and materials. The process of reducing the data to obtain an inflation rate for the many time periods included in this thesis would prove extremely cumbersome. The accuracy would still remain questionable due to the time spans involved in ship construction (i.e. present labor costs and materials bought two years previously). In addition, wage rates vary from coast to coast and the Wholesale Price Indices are approximations to construction materials utilized in the Weight Groups listed. Furthermore, the relationship between shipyard costs and prices will depend on the overall supply/demand relationship for the U.S. shipbuilding industry at the time of the construction bids. The simplified method of determining the inflation rate is considered a reasonable engineering solution. This inflation rate is reasonable and is selected for use in bringing all cost estimates to the base year of 1976.

The Yuhas estimation technique is applied to the POLAR STAR, M1 (conventional), and M2 (nuclear). The results are presented in Table 5.1. The POLAR STAR results are provided for reference and validation of the technique. The U.S.C.G. calculated an approximate Contract Cost Estimate of 58 million dollars for the POLAR STAR, based on 1972 dollars [47]. The actual cost of the POLAR STAR, to date, assuming all claims settled

Table 5.1

YUHAS COST ESTIMATION

Weight Group		Relationship Ship Units	POLAR STAR ($\$X10^6$)	M1 ($\$X10^6$)	M2 ($\$X10^6$)
W_1	M	\$280/T	1.400	2.367	1.595
	L	350 MH/T	6.125	10.356	6.98
W_2	M	Diesel Plant- \$90/SHP	5.4	2.97	2.7
	*M	Remainder \$2,000/T	3.214	2.838	5.34
	L	200 MH/T	1.125	.993	1.869
W_3	M	\$5,500/T	1.04	1.237	1.332
	L	650 MH/T	.43	.512	.551
W_4	**M	\$5000/T	.125	.125	.125
	L	1200 MH/T	.105	.105	.105
W_5	M	\$3000/T	2.25	3.807	2.335
	L	400 MH/T	1.05	1.777	1.089
W_6	M	\$5300/T	3.355	5.671	3.354
	L	400 MH/T	.886	1.498	.886
W_7	CLASSIFIED		UNKNOWN	UNKNOWN	UNKNOWN
W_8	M	3% TMC for W_1-W_7	.504	.570	.503
	L	10% TLC for W_1-W_7	.972	1.524	1.148
W_9	M	3% TMC	.504	.570	.503
	L	10% TLC	.972	1.524	1.148
Total	M		17.792	20.155	17.788
	L		11.665	18.288	13.776
Overhead - 70% of Total L			8.166	12.802	9.643
Profit - 10% of Total L+M+ overhead			3.762	5.124	4.121
Contract Cost Estimate Total L+M+Overhead + Profit			41.385	56.370	45.328
Inflation rate 7.3%/year(1972)			58.863	80.176	64.471
(1976)			78.026	106.277	85.460
Corrected 1976 Contract Cost Estimate			70.223	95.650	76.914
Final End Cost Estimate			84.268	114.780	92.297

Table 5.1 (cont.)

Notes: T = Tons, MH/T = Man-Hours/Ton

* Total weight was used as a conservative approximation, W_2 relationships do not include electric propulsion, only the diesel is included by Yuhas.

** This figure includes GFM weight as an approximation. Yuhas does not include GFM weight.

TMC - Total Material Cost

TLC - Total Labor Cost

in favor of the builder, is $52,865,000 [47]. Lockheed won the original contract with a bid of $52,681,485 [45]. Possibly, modern ship building has managed to reduce costs. This could account for the eleven percent overestimation provided by Yuhas. The Contract Cost Estimates in Tabel 5.1 are, therefore, reduced ten percent.

The results in Table 5.1 appear to be reasonable. The significantly larger conventional vessel (M1) is more costly than the smaller nuclear vessel (M2). M2 is more costly than the POLAR STAR, even though they share identical hull forms. This is expected as a result of the higher nuclear ship density. The reader must be reminded that the Yuhas approach is based on light ship weights.

M2 (nuclear) should be more expensive than the Yuhas method indicates. Yuhas is diesel plant oriented. This increase would be reflected in the cost per ton, and in the required man-hours per ton. Increased complexity, special materials, and multiple safety checks are just a few items causing higher costs. The ratio of M2 Contract Cost Estimate to POLAR STAR Contract Cost Estimate indicates a 10 percent cost increase due to nuclear propulsion. Based on [48], a 25 to 35 percent cost increase due to nuclear propulsion is expected. The M2 Contract Cost Estimate would, therefore, be in the $87 and $94 million (1976) range.

In addition to the steps presented in Table 5.1 for deriving Contract Cost Estimate, Yuhas provides relationships for obtaining a Final End Cost Estimate (Table 5.2). Insufficient data from the POLAR STAR prohibit detailed validation of this section of the Yuhas method. The author will apply a 20 percent increase to the Contract Cost Estimate to obtain the Final End Cost Estimate.

The remaining conventional icebreaker cost data to be investigated are from the NUS study [11]. This data addressed the procurement, construction, and operating costs of the various propulsion plants. The entire ship costs were not investigated by [11]. The NUS study is utilized primarily to provide diesel electric propulsion plant validity to Yuhas, and to provide operating cost estimates.

The cost tabulation for Diesel Electric propulsion, performed in the NUS study, is segmented by Weight Groups (W_2, W_3, W_5). The costs for W_3 and W_5 are not the total W_3 and W_5 values, but only the propulsion plant related costs. The W_2 cost, for a Diesel Electric plant of 35,000 SHP, ship with a cubic number equal to 16.8, W_2 of 1580 Tons, and Ft_{LBP} of 400, (1966 dollars) is $6,142,650. The cost of W_2 in 1967 dollars is $6,591,063. The W_2 Yuhas cost for M1 (SHP = 33,000, Ft_{LBP} = 419, W_2 = 1419 Tons, CN = 20.9) is $6,801,000 (Table 5.1). This agreement serves to validate the Yuhas propulsion machinery cost estimate. The Yuhas method will provide the procurement

Table 5.2

YUHAS CONTRACT TO FINAL END COST

a.	Basic Change Orders	(12% owner's basic construction cost (OBC) OBC = SBC + Profit
b.	Escalation and Growth (3 year lag time)	(7.5% OBC)
c.	Post Delivery	(3% OBC)
d.	GFM Costs	(From Shopping List)
e.	Future Characteristics Changes	(2.5% OBC + totals of Items <u>a</u> thru <u>d</u> above)
f.	Extra Repair Parts	(From Shopping List)
g.	Long Lead Procurement Items	(Man-Hour Cost + Material costs escalation factor)
h.	Coast Guard Resident Inspectors	(Man-Year Costs)
i.	Coast Guard Administrators and Designers	(Man-Year Costs)
j.	Shore-based Spares	(From Shopping List)
k.	Expendable Materials (china, mattresses, etc.)	(From Shopping List)

and construction costs for the conventional Diesel Electric ship system.

5.1.2 Operational costs

Operating costs for a Diesel Electric propulsion plant are composed of several items: fuel, maintenance and repairs, manning, lube oil, etc.

5.1.2.1 Fuel oil costs

Fuel costs will generally provide the most significant operational cost differential between conventional and nuclear plants. Determination of fuel cost per year obviously requires fuel consumption per year. Fuel consumption is a function of operating profile. A micro-profile (one mission) and resultant fuel consumption (8702 Tons) for M1 are provided in Table 3.6. However, a macro-profile of several years, including a yard period, is necessary to determine a reasonable annual fuel consumption. An expected average annual deployment time of 6-7 months is specified in the original task statement [1]. This thesis will use 200 days (6.57 months) underway, or approximately 2.6 seventy-seven day missions, as the operating profile. The average annual cost of fuel oil for the icebreaker, (M1), powered by a fossil propulsion plant, is the consumption of oil (barrels) times the cost of oil (dollars/barrel). The cost of Marine 2 diesel fuel for March to August 1976 is $13.50/barrel [49].

Consumption per 77 day mission = 8702 tons

Cost of fuel oil - \$13.50/barrel

One ton diesel oil - 7.391 barrels [32]

$\therefore$ Average Annual Fuel Oil Cost

= 2.6 missions (8702 tons/mission)(7.391 barrels/ton) (\$13.50 barrel)

= \$2.258 million/year

5.1.2.2 Maintenance and repair costs

Maintenance and repair costs (MRC) are the next operational costs to be investigated. The value for MRC presented by [11] for a 35,000 SHP icebreaker is \$136,580 (\$260,513 in 1976 dollars). This value is based on a 169 days per year deployment and power utilization profile [11]; (M1 mission profile contained in Table 3.5):

35.5% at 100% power

17.7% at 80% power

17.7% at 50% power

29.1% at 40% power

Femenia [29] presents the following general equation (1973 dollars):

$$MRC = 9.4 \left(\frac{SHP}{1000}\right) \text{seadays} + 4875 \left(\frac{SHP}{1000}\right)^{2/3}$$

$$= 9.4\ (33)\ 200 + 4875\ (33)^{2/3}$$

$$= \$112,195$$

MRC (1976) = \$138,603

Femenia assumes a diesel plant with reduction gear-driven main generators and two diesel-driven auxiliary generators. Femenia does not deal with icebreakers and because of the severe operating conditions, the MRC of an icebreaker should be higher. The values for the two MRCs validate the order of magnitude. The NUS data will be utilized for M1.

5.1.2.3 Manning costs

Personnel costs for manning an icebreaker contribute a significant portion of the annual operating costs. A rough personnel profile [50] for the POLAR STAR is presented in Table 5.3.

The M1 complement is assumed to be a linear function of cubic number. Figure 5.1 contains a plot of complement versus cubic number. The data are taken from [11]. The following linear equation was formed:

$$Y = 13.45X + 224.89$$

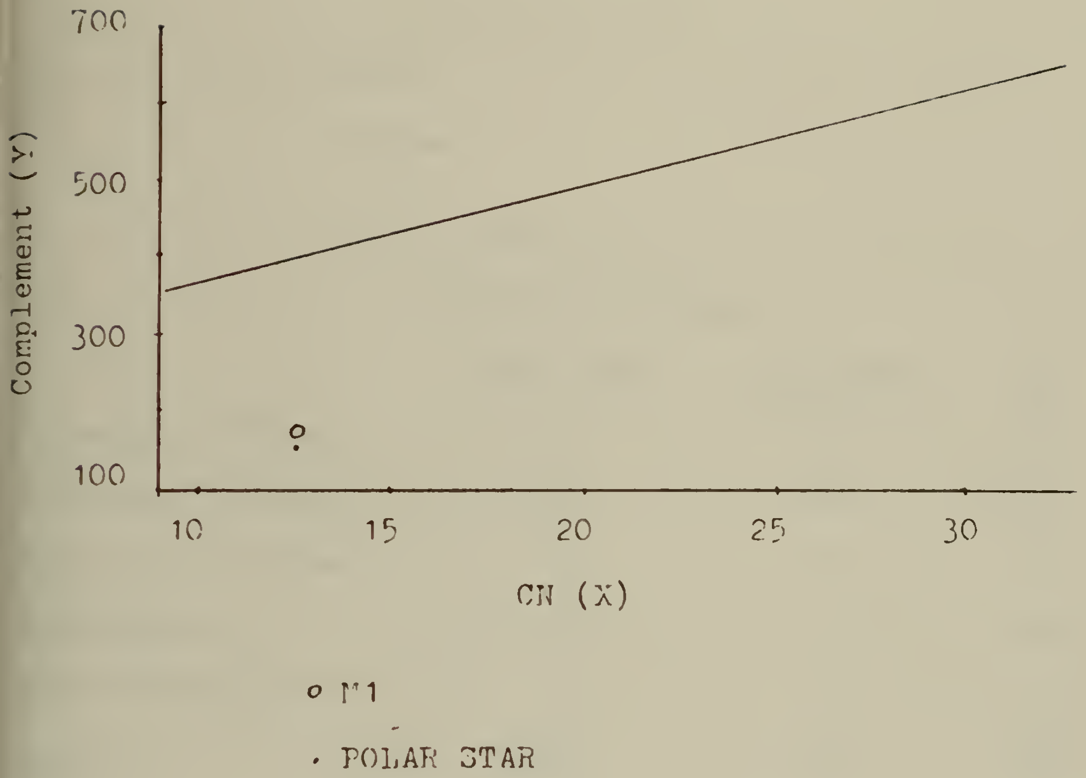
Obviously, the POLAR STAR complement does not fall on this line (Figure 5.1) reflecting a change in U.S.C.G. manning philosophy. However, the linear assumption and the slope will be maintained in determining a model relationship for M1. The resultant complement is apportioned in the same percentages as the POLAR STAR (Table 5.3).

$$S = 13.45 R - 2.24$$

where:

Figure 5.1

Manning Relationship



$$R = \text{cubic number}$$

$$S = \text{complement}$$

$$\therefore S[M1] = 13.45\,R\,[M1] - 2.24$$

$$= 13.45\,(20.9) - 2.24$$

$$= 278.9$$

$$= 279 \text{ men}$$

Table 5.3

COMPLEMENT PROFILE

	POLAR STAR	% of Total	M1	M2
Permanent Officers	13	8	22	14
Permanent Enlisted	125	76	212	129
Aviation Detachment	15	9	25	15
Scientists	10	6	17	10
Medical Officer	1	1	3	2
Totals	164	100	279	170

The average annual manning cost will be computed from average compensation figures published in a November 1975 Navy Times. Compensation includes base salary, allowances, and an approximation for medical, tax, exchange, and commissary benefits.

For the sake of simplicity and without any significant loss of accuracy in this phase of conceptual design, the average officers' compensation will be represented by a Lieutenant (0-3), the enlisted complement by a second class

petty officer (E-5). The scientists will be considered officers. Four men of the aviation detachment on the POLAR STAR (25%) are pilots and will be added to the officer complement, the remainder added to the enlisted complement. Table 5.4 illustrates the annual cost calculation.

5.1.2.4 Lube oil cost

Lube oil is the final operating cost that will be investigated. This can be a significant cost to a diesel propulsion plant. The NUS study and Femenia will be employed to determine average annual lube oil costs.

Reference [11] determines the lube oil cost (1966 dollars) for main engines, diesel generators, and reduction gears to be $52,500. This value is based on the 35,000 SHP vessel and the mission profile presented in the MRC section. Lube oil consumption is dependent on shaft horsepower and running time. The NUS ship is underway only 169 days versus 200 days for M1. However, the power level utilization is higher for the NUS study. These are assumed to be offsetting differences. Only an adjustment for SHP will be made. Lube oil consumption is assumed to be linear with SHP over a small range. Therefore, an estimated lube oil cost per year for M1 in 1976 dollars is determined as follows:

$$\$52{,}500 \; \frac{33{,}000 \text{ SHP}}{35{,}000 \text{ SHP}} = \$49{,}500 \text{ (1966 dollars)}$$

Table 5.4

ANNUAL MANNING COST

	Compensation	POLAR STAR	M1	M2
Permanent Officers	24,000	312,000	528,000	336,000
Aviation Detachment	24,000	96,000	144,000	96,000
Scientists	24,000	240,000	408,000	204,000
Medical Officer	24,000	24,000	72,000	48,000
Sub Total		672,000	1,152,000	720,000
Permanent Enlisted	12,995	1,624,375	2,754,940	1,676,355
Aviation Detachment	12,995	142,945	246,905	142,945
Sub Total		1,767,320	3,001,845	1,819,300
Total ($/year)		2,439,320	4,153,845	2,539,300

$$\$49,500\left(\frac{\$1.90\ (1976)}{\$1.14\ (1966)}\right) = \$82,500\ (1976\ \text{dollars})$$

Femenia provides equations for medium speed main diesel engine and auxiliary diesel generator lube oil consumption.

Annual L.O. Cost = MEC(HM) CMLO + APC(HA) CALO

where:

MEC = Main Engine Consumption

= 2.42×10^{-3} lb/BHP-HR

HM = Hours of main engine operation

CMLO = Cost of main engine lube oil (Military Grade 9250)

= \$1.90/gallon (1976)

APC = Auxiliary Engine Consumption

= 3.07×10^{-3} lb/HP-HR

HA = Hours of auxiliary engine operation

CALO = Cost of auxiliary engine lube oil

= \$1.90/gallon (1976 dollars)

HM (from Table 3.5) = 48 full power days per 77-day mission

= 48/77 (200)

= 124.7 full power days per year

= 2992.8 hours/year

KW for M1 = $N + 6.3\sqrt{N}$

= $279 + 6.3\sqrt{279}$

= 384

Select 500 KW, allowing approximately 20% growth margin

HP = 1.341 (500 KW) = 670.5 HP

Assume .8 load factor underway and electrical shore tie in port.

HA = (.8) 200 days = 160 days/year = 3840 hrs/year

$$\text{MEC (Main)} = \frac{1\ \text{Ton}}{2240\ \text{lb}}\ \frac{42.42\ \text{ft}^3}{\text{Ton}}\ \frac{7.48\ \text{gal}}{\text{ft}^3}\ \frac{2.42 \times 10^{-3}\ \text{lb}}{\text{BHP - HR}}\ 33{,}000\ \text{BHP}$$

$$= 10.98\ \text{gal/hr}$$

$$\text{APC (Aux.)} = \frac{1\ \text{Ton}}{2240\ \text{lb}}\ \frac{42.42\ \text{ft}^3}{\text{Ton}}\ \frac{7.48\ \text{gal}}{\text{ft}^3}\ \frac{3.07 \times 10^{-3}\ \text{lb}}{\text{BHP - HR}}\ 670.5\ \text{BHP}$$

$$= .292\ \text{gal/hr}$$

Annual L.O. Cost = 10.98 gal/hr (2992.8 hr/yr) \$1.90/gal

+ .292 gal/hr (3840 hr/yr) \$1.90/gal

= 41,733 + 1424

= \$64,566/year (1976 dollars)

The reader is reminded that Femenia's values do not include reduction gear lube oil, nor are they representative of icebreakers. The value selected for use in this thesis is $100,000 per year.

The direct cost analysis for the Diesel Electric powered M1 is complete. A summary of costs in 1976 dollars is contained in Table 5.5.

Table 5.5

CONVENTIONAL PLANT COSTS

Capital Cost	M1
Final End Cost Estimate (5.1)	114,780,000
Operating Costs ($/year)	
Fuel Oil (5.1.2.1)	2,258,000
Maintenance and Repair (5.1.2.2)	260,513
Manning (5.1.2.3)	4,153,845
Lube oil (5.1.2.4)	100,000
Total	6,772,358

5.2 Cost estimation for nuclear systems

5.2.1 Capital costs

5.2.1.1 Ship costs

Due to the relative scarcity of data, nuclear ship cost estimation is in its infancy. Some attempts have been made to deal with tankers or container ships [36, 51, 52]. A study of nuclear power application to polar icebreakers was performed for the U.S.C.G. in 1966 [11]. More recently (1973), Global Marine Engineering Company performed a feasibility study of a nuclear powerd icebreaking support ship [37]. The lack of applicable data will necessitate some assumptions with little or no verification.

Caroussis [36] presents estimating relationships for costs dealing with: power plant procurement and installation, ship construction, and operating costs. Many of the nuclear power plant relationships deal with the plant selected for M2, the Babcock and Wilcox Consolidated Nuclear Steam Generator. However, the plant is employed in a tanker application. Calculation of the nuclear propulsion package cost (C_{MN}) is broken into several segments:

1. C_{NNM} = non-nuclear machinery (including installation)

$$= \$1,112,500\ (K)(SHP/1000)^{0.487}$$

K = 1.274 Triple screw, aft

$$C_{NNM} = 1,112,500\ (1.274)(30)^{0.487}$$

$$= \$7,427,241$$

2. C_{NSSS} = nuclear steam supply system (without installation, includes a 10% miscellaneous materials margin)

$= 4{,}774{,}000 + 62.15\ (SHP)$

$= \$6{,}638{,}500$

3. C_{NSSSE} = nuclear supply system installation (fairly independent of size)

$= \$1{,}570{,}000$ (CNSG III)

4. C_{SC} = containment and shielding (including installation)

$= \$826{,}300 + \$13{,}750\ (SHP/1000)^{.965}$ (CNSG III)

$= \$1{,}192{,}505$

5. C_{OHC} = miscellaneous costs

Start up and test	1,180,000
Collision barrier	500,000
Initial crew training	420,000
Other hull changes and additions	1,250,000
Licensing fee	350,000
Spare parts	800,000
	4,500,000

6. $C_{NM} = C_{NNM} + C_{NSSS} + C_{NSSSE} + C_{SC} + C_{OHC}$

$= 7{,}427{,}241 + 6{,}638{,}500 + 1{,}570{,}000 + 1{,}192{,}505 + 4{,}500{,}000$

$= \$21{,}328{,}246$ (1974 dollars)

$= \$24{,}555{,}828$ (1976 dollars)

Note: Included in C_{NNM} and C_{SC} are a 10% margin for miscellaneous labor and an overhead rate of 70%.

Construction cost for the entire vessel is addressed in [36]. The calculation for M2 is contained in Appendix E.

Comparison of $69,843,314 from Caroussis (Appendix E) with the Yuhas cost estimations for the POLAR STAR, M1, and M2 presented in Table 5.1, indicates that the Caroussis value for the total ship is too low. However, if the Yuhas cost values for W_2 (from M2) are isolated and run through the last steps of the method (W_8, W_9, Overhead, not Profit, and inflated to 1976 dollars), the value derived, $23.27 million, compares well with C_{NM} ($24.56 million) from Caroussis. The close agreement in propulsion plant costs tends to add validity to the Yuhas weight-oriented method. Since the costs of the two methods diverge after the propulsion plant determination, the lack of applicability of containership-based relationships to polar icebreakers becomes readily apparent.

The NUS study was investigated next to attempt to validate the cost of a nuclear propulsion plant. The steam supply system includes a loop-type reactor, not the more compact and integral CNSG unit. The cost of the steam supply system for a 400 foot (Ft_{LBP}), 35,000 SHP, and cubic number of 16.8, vessel is $7,055,000 (1966 dollars) or $14,272,309 (1976 dollars). This value includes the reactor, associated auxiliary systems, outfit for nuclear labs, a 25% manufacturer's markup, and

spare parts. The total W_2 cost in 1966 dollars is $13,740,350 or $27,796,814 in 1976 dollars. The 13 percent increase over the tanker formulation is expected due to more stringent operating conditions as well as inclusion of more safety equipment.

The Global Marine Engineering Company investigated nuclear power application to an icebreaking support ship for the Arctic [37]. The proposed ship is comparable in external dimensions with those mentioned previously in this thesis.

LOA	418'
LBP	381'9"
CN	10.44
SHP	24,300
BWL	76'
T	26'
Δ at T	14395 Tons

The Global Marine study presents no detailed data or method for determining the Reactor or Ship Construction Costs. Values for Ship Construction of $96 million and a Reactor cost of $50 million (including $4.6 million for four-year fuel cost) are given. After subtracting the fuel cost and converting to 1976 dollars, the total cost of the ship is $174,682,169. This value appears to be high. Without a detailed cost breakdown, it is difficult to speculate why the cost of this vessel is so high. However, this supply ship is outfitted with: a 10,000 HP Pneumatically Induced Pitching System, (including four 2500 HP motors), two 45 ton diesel powered cargo handling

cranes, a 113.5 Mwt reactor vs. 95 Mwt (M2), and other supply-oriented equipment. The Global Marine study provides upper limits for the cost of M2.

A summary of nuclear plant and total ship costs (1976 dollars) are presented in Table 5.6. The first column is based on an expected 25 to 35 percent increase [48] in cost of nuclear over conventional as a rough estimate. The Yuhas figure, Final End Cost Estimate, for the POLAR STAR was increased by 25 to 35 percent.

The modified M2 (M2A) was derived by dropping the W_2 cost from Yuhas and adding the more applicable NUS Weight Group 2 cost.

NUS

W_2 (1966) = 13.74 million

Yuhas

W_2 (1967) = 9.909 million

Assumptions

1. Yuhas (M2) and NUS Weight Group 2 are approximately compatible.
2. Ratio of W_2 Labor Cost to W_2 total for Yuhas is maintained in NUS data.

$$\text{Labor [NUS]} = \frac{1.869\text{M}}{9.909\text{M}}(13.74\text{M}) = 2.592\text{M}$$

$$\text{Material [NUS]} = 13.74 - 2.592 = 11.148\text{M}$$

Employ Yuhas as in Table 5.1.

Total	M	20.896
	L	14.499
Overhead--70% of Total L		10.149
Profit--10% of Total L + M + Overhead		4.55
Contract Cost Estimate		50.099
Inflation to 1976 dollars		94.455
Corrected 1976 Cost (10%)		85.009
Final End Cost Estimate for M2A		102.011

The average of the values for Ship Cost was found to be approximately $110 million. This represents a cost increase of approximately 30% for nuclear versus conventional power plants.

Table 5.6

NUCLEAR SHIP COST COMPARISON

	[45]	Yuhas M2	Yuhas M2A	Caroussis	NUS	Global
Plant Cost	--	23.27	27.8	24.56	27.8	45.6
Ship Cost	105-113	92.30	102.01	69.84	--	174.68

Costs are in millions, 1976 dollars, 7.3% inflation rate per year.

The cost of procurement and construction for the Nuclear-Steam Turbo-Electric ship system is selected as $110 million.

5.2.1.2 Shore facilities

A capital cost above conventional ship cost is involved in shore facilities when considering a nuclear power plant. Shore facilities include ability to perform

reactor maintenance, refuel, and handling of contaminated reactor components. NUS includes establishment of a Training Staff and Technical Support Team at a cost of $1,000,000 (1976 dollars).

The capital cost of the shore facilities is difficult to derive. The cost per vessel is dramatically reduced if spread out over several vessels. With present developing MARAD interest and U.S. Navy expertise in nuclear power, it is logical to assume that common facilities could be formulated, cutting costs to all users.

This thesis will assume an initial shore facilities cost of $1,500,000 (1976). The NUS study computes a cost of $1,460,000 (1976). Caroussis calculates a cost of $1,800,000 (1976), including tanker pier and other related costs.

5.2.2 Operational costs

Operating costs for a Nuclear-Steam Turbo-Electric propulsion plant are composed of several items: fuel, maintenance and repairs, manning, etc.

5.2.2.1 Fuel costs

The determination of nuclear fuel costs is a complex procedure, requiring analysis of the fuel cycle and resultant cash flows. Appendix F contains the necessary equations, parameters, and assumptions. The annual fuel

cost is a function of the discount rate selected. As a result, nuclear fuel cycle costing is performed in Section 5.3.

The basic refueling cycle for the reactor is taken to be 10 years. This is in accordance with U.S.C.G. desires presented in [11]. Clearly, this assumes no change in U.S.C.G. policy. This is obtained with 4.7% U^{235} enrichment and an acceptable 19,097 Megawatt-Days/ Tonne (Mwd/Tonne) reactor burnup.

5.2.2.2 Maintenance and repair costs

Nuclear MRC are presented by Caroussis [36] and the NUS study [11]. Caroussis estimates $150,000 (1974 dollars) for the nuclear plant based on a 300 days per year operating profile. This is the equivalent to $172,700 in 1976 dollars.

The NUS study estimates the annual MRC in 1966 dollars to be $97,000 (for a 35,000 SHP ship). Converted to 1976 dollars, MRC equals $196,232. The operating profile is contained in Section 5.1.2.2.

A conservative estimate of $200,000 (1976 dollars) will be considered appropriate for the M2.

5.2.2.3 Manning costs

Manning costs for M2 were computed as in Section 5.1.2.3. Table 5.3 presents a possible personnel profile. The line plotted in Figure 5.1 was utilized to form a

relationship for determining the complement. Linearity and slope were assumed to remain valid.

$V = 13.45U + 3.76$

where:

U = cubic number

V = complement

The complement for M2 was established at 170, assuming six additional men were sufficient to handle a nuclear plant. Six were considered adequate in the NUS study. The resulting annual cost for M2 is contained in Table 5.4 ($2,539,300).

5.2.2.4 Consumable supply costs

Consumable supply costs per year are estimated by NUS as $22,000 (1966) or $44,500 (1976). The cost value includes boiler chemicals, ion exchange resins, nitrogen for containment inerting, protective clothing, film badges, etc. An operating cost of $45,000 is selected as the annual consumable supply cost.

5.2.2.5 Shore facilities costs

Operation of a nuclear vessel requires technical support and training staffs. The incremental impact on the present U.S.C.G. technical and staffs is very difficult to assess. The U.S. Navy has established expertise and training facilities. Suggested costs from [11] of $400,000 (1976) and [36] of $500,000 (1976), are considered

Table 5.7

NUCLEAR PLANT COSTS

	M2
Capital Cost	
Final End Cost Estimate (5.2.1.1)	110,000,000
Shore Facilities (5.2.1.2)	1,500,000
Total	111,500,000
Operating Costs ($/year)	
Fuel (5.2.2.1)	
Maintenance and Repair (5.2.2.2)	200,000
Manning (5.2.2.3)	2,539,300
Consumable Supplies (5.2.2.4)	45,000
Shore Facilities (5.2.2.5)	500,000
Total	

reasonable, under present circumstances. These costs are subject to reductions as fleet size increases. The annual shore facility operating cost is selected as $500,000 (1976).

The direct cost analysis for the Nuclear-Steam Turbo-Electric powered M2 is complete. A summary of costs (1976 dollars) is displayed in Table 5.7.

5.3 Economic comparison

The objective of this section is to perform an economic comparison of the two ships, M1 and M2. The capital costs and annual operating costs developed for each of these vessels in previous sections will provide the necessary data. The discounted cash flow or present value technique will be employed to analyze the various

cash flows. The author is not going to discuss the advantages and pitfalls of present value versus payback, average rate of return, or internal rate or return. Any reader not familiar with these capital budgeting techniques is referred to any good financial management text, such as Van Horne [53].

Utilization of present value techniques requires the selection of a discount rate. This should ideally be the cost of capital to the investor or buyer. This raises the question of: What is the U.S.C.G.'s cost of capital?

The U.S.C.G. is a nonprofit, non-revenue earning, government agency. Does the government provide an endless supply of funds, implying no cost of capital? Definitely not! The supply of funds is certainly limited. Money for a U.S.C.G. icebreaker must come at the sacrifice of another government project or agency, or by raising the taxes of the country's people. At this point, the discussion leads to utility. The benefit from an icebreaker must be weighed against welfare checks, a submarine, a moon shot, or raising a bus driver's taxes. Each of these utilities implies a cost of capital to the government and U.S.C.G. In view of the difficulty involved in determining a discount rate, two rates will be selected for the analysis (10% and 20%).

A summary of propulsion plant costs for M1 and M2 is contained in Table 5.8. The M1 data has been taken from

Table 5.8

SUMMARY OF SHIP COSTS

	10%			20%		
	M1	M2	P.S.	M1	M2	P.S.
Capital Cost	114.780	111.500	84.268	114.780	111.500	84.268
Final Decon.		.011			.001	
	114.780	111.511	84.268	114.780	111.501	84.268
Operating Costs ($/year)						
Fuel	2.258	.756	1.826	2.258	.871	1.826
M and R	.260	.200	.237	.260	.200	.237
Manning	4.154	2.539	2.439	4.154	2.539	2.439
Lube Oil	.100	-------	.091	.100	-------	.091
Consumables	-------	.045	------	-------	.045	------
Shore Facilities	-------	.500	------	-------	.500	------
EACC	12.182	11.835	8.944	23.053	22.394	16.925
Total EACF	18.954	15.875	13.537	29.825	26.549	21.518

Notes: All costs are in millions

EACC = Equivalent Annual Capital Cost

EACF = Equivalent Annual Cash Flow

Table 5.5, M2 from Table 5.7. All capital costs have been converted to an Equivalent Annual Cash Flow (EACF) and added to the annual operating costs.

EACF = PVCC [r/(1 - R exp (-LE))]

where:

PVCC = Present Value Capital Cost

r = discount rate

R = 1 + r

LE = economic life of ship

Sample for M1

PVCC = 114,780,000

r = 10%

$$EACF = 114,780,000 \left[\frac{.1}{1 - (1.1)^{-30}}\right]$$

$$= 114,780,000\ (.106) = \$12,182,182/\text{year}$$

For simplicity, the ships have been assigned a salvage value of zero. The nuclear ship (M2) will have a final decontamination cost of $200,000 [11]. This has been discounted over 30 years and added to the capital cost. The final nuclear fuel credit has been taken into account in the fuel calculations.

Investigation of Table 5.8 leaves no doubt as to the optimum economic choice between M1 or M2. The nuclear powered ship (M2) has an Equivalent Annual Cash Flow approximately nineteen percent (19%) less than the fossil fueled ship (M1).

The first significant cost savings experienced by M2 results from the annual fuel cost. Therefore, a sensitivity analysis of fuel oil prices (± 20%) was performed. Values of AECF were recalculated. The results are presented in Table 5.9 and plotted in Figures 5.2 and 5.3, for 10 and 20 percent discount rates respectively. M2 shows no sensitivity to fuel oil prices. Figure 5.2 indicates that M1 would become an economically equal choice with M2 at a fuel price of -$6 per barrel (144% decrease). At a discount rate of 20%, Figure 5.3 yields an equivalence point at -$7 per barrel (152% decrease). As the fuel price increases M2 becomes a better selection.

The remaining and largest cost saving afforded M2, results from the huge increase in manning costs charged to M1. Manning costs are shown in Table 5.4. The complement of M1 (279) shows an increase over M2 (170) of 64 percent. However, these ships perform the identical mission. The reader is also reminded that M1 constitutes an increase in displacement solely to accommodate fuel oil. Additional tankage certainly does not require a 64 percent complement increase. This apparent incongruity warrants a re-evaluation of the M1 personnel requirements.

The linear equation developed in Section 5.1.2.3 expresses complement as a function of cubic number. M1 would appear to be a special case where the equation is not

Table 5.9

FUEL OIL SENSITIVITY ANALYSIS

%	Fuel Price		r = 10% FUEL (M)	EACF (M)	r = 20% FUEL (M)	EACF (M)
+20	16.20	M1	2.710	19.407	2.710	30.277
		M2	.756	15.876	.871	26.549
		PS	2.463	14.174	2.463	22.155
0	13.50	M1	2.258	18.955	2.258	29.825
		M2	.756	15.876	.871	26.549
		PS	2.052	13.763	2.052	21.744
-20	10.80	M1	1.806	18.503	1.806	29.374
		M2	.756	15.876	.871	26.549
		PS	1.642	13.353	1.642	21.334

M = millions

Figure 5.2

Fuel Price Sensitivity
r = 10%

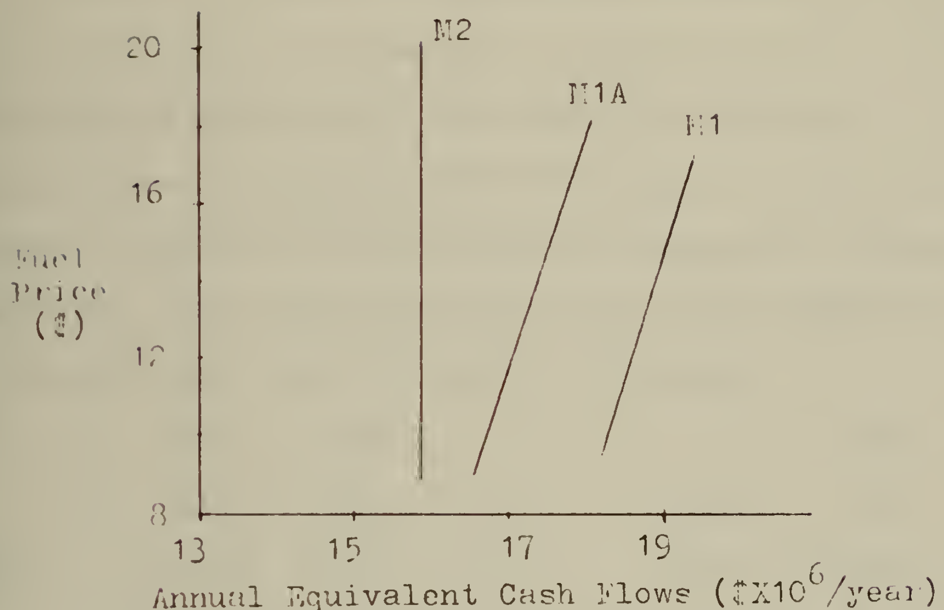
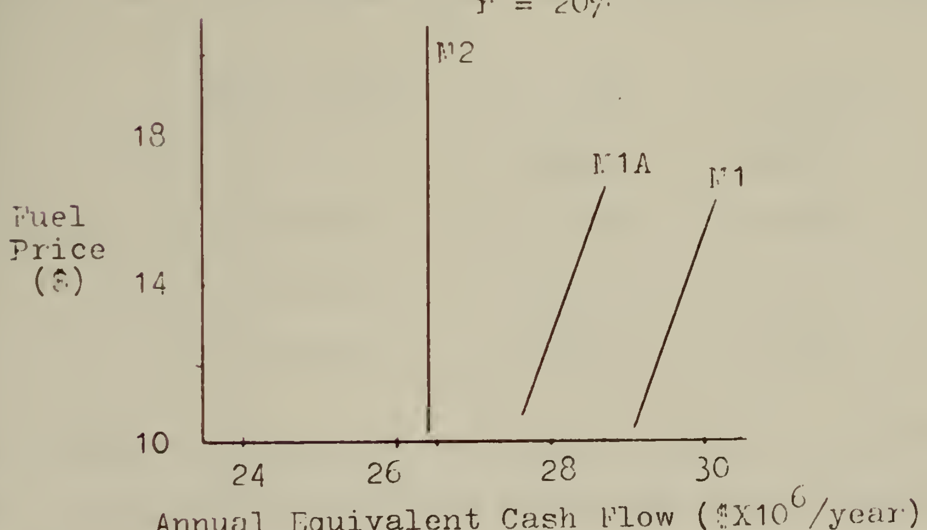


Figure 5.3

Fuel Price Sensitivity
r = 20%



applicable. Obviously, the 21,730 ton M1 will require a larger complement than the 12,862 ton M2, but how much? To determine the EACF's sensitivity to manning, the limiting case was chosen. The change in manning from M2 to M1 was assumed to be zero. The output of new EACF calculations, assuming present fuel price of $13.50 per barrel, indicate that M2 remains the superior economic selection. At a discount rate of 10%, M2 commands a 9.2% advantage, and at 20%, a 6.3% margin.

	10%			20%		
	M1	M2	PS	M1	M2	PS
EACF	17.340	15.874	13.763	28.210	26.549	21.744

Now that both major contributors to the M2 cost advantage have been evaluated independently, it seems logical to investigate the effect of their combination. Manning was assumed equal for M1 and M2. The M1 values of EACF were recomputed over the ± 20% fuel oil price range.

Fuel Price	r = .10	r = .20
	EACF	EACF
$16.20	17.792	28.662
$13.50	17.340	28.210
$10.80	16.888	27.759

The results are plotted on Figure 5.2 and 5.3, labeled as M1A. A shift in fuel price to $5 per barrel (63% decrease) would cause the EACF of M1A and M2 to be equal, when subject

to 10% cost of capital. At 20%, fuel price must decrease 78% to $3 per barrel.

Concluding this section entails summarizing comments concerning the comparison of M1 - M2. The goal of this chapter was to identify an economic optimum, choosing M1 or M2. It must be kept in mind that a conservative approach was utilized in regard to M2. Selecting a severe set of circumstances; a 20% discount, a 20% drop in fuel oil price, and no extra manning costs charged to M1, the nuclear vessel (M2) still maintains a 5% advantage (26.549 versus 27.759 million dollars) in Equivalent Annual Cash Flows. As the situation swings in the favorable direction, 10% discount, full manning charge, and 20% increase in fuel oil price, the advantage of M2 increases to 22%. M2 is the desired economic optimum choice.

An interesting tangent, at this point, is to perform a brief comparison of M2 and the POLAR STAR. The POLAR STAR and M2 share the same hull form. This is where the similarity ends. The POLAR STAR has a 28.5 full power day mission endurance (vs. 48 days). The POLAR STAR is powered by a combination diesel (18,000 SHP) or gas turbine (60,000 SHP) electric plant. This CODOG propulsion plant makes application of diesel electric (30,000 SHP) powered M1 relationships invalid. By economic necessity, the POLAR STAR mission length and structure must be different from that of M1 (Table 3.5). Even at 50% power the POLAR STAR would require utilization of two inefficient gas turbines.

The following assumptions are made to facilitate determination of operational costs of the POLAR STAR.

1. Fuel consumption per mission is determined by assuming that 95% (tailpipe allowance) of the ship's total fuel allowance is burned every trip (4160 Tons).
2. Operation for 200 days per year
3. Even though structure of mission profile must be different than M1, let the following ratios be equal:

   28.5 full power days/48 full power days = x/77 days

   x = 45.7 days per mission

   ∴ 200 days/45.7 days = 4.4 missions per year
4. Maintenance and Repair Costs are approximately equal to M1.
5. Lube Oil Costs equal to M1. Less diesel oil is consumed, but the more expensive synthetic turbine compensates.

Utilizing the POLAR STAR capital cost calculation (Table 5.1), manning cost (Table 5.4), fuel cost calculation as per Section 5.1.2.1 (4160 Tons and 4.4 missions per as inputs), and assumptions 4 and 5, the EACF was determined (Table 5.8). The POLAR STAR is approximately 17% less expensive to operate at a 10% discount rate, 23% at 20% discount rate, than M2. Before jumping to conclusions, it must be noted that M2 has a 68% advantage

in length of mission. Even though both vessels are underway 200 days per year, the POLAR STAR time in ice and/or cruising radius are sharply reduced. An obvious tradeoff is apparent: a 17% increase in costs for a 68% longer mission endurance. This is the problem facing the decisionmakers in Headquarters. The author does not attempt to pose a solution. It must be pointed out that when the POLAR STAR was under study, the fuel oil price was $4.20 per barrel.

5.4 Nuclear implications

Chapter V has demonstrated the economic desirability of the Nuclear-Steam Turbo-Electric powered M2 ship system. There are additional factors that should play a role in the decision to procure and operate a nuclear icebreaker.

--Endurance capabilities in excess of mission requirements. The propulsion system no longer limits the ship's endurance.

--Training of U.S.C.G. personnel to support and operate nuclear icebreakers. Impact could be minimized by utilization of U.S. Navy, MARAD, and other government agencies' facilities and expertise.

--Influx of high calibre personnel into a nuclear program may deplete other U.S.C.G. disciplines.

--Jurisdictional conflicts between U.S.C.G. and other governmental agencies, such as Atomic Energy Commission, Nuclear Regulatory Commission, and the Environmental Protection Agency.

--Political parameters. Lack of international nuclear policy governing safety requirements, licensing practices, foreign port costs, accident insurance, etc. The Convention on Liability of Operators of Nuclear Ships adopted in 1962 but not enacted, at least demonstrates recognition of a problem [45].

These items are not meant to be all-inclusive. They are included to provide a suggestion of non-economic elements, which should be considered along with Equivalent Annual Cash Flows, when evaluating nuclear power.

The objectives of this chapter have been achieved. Conventional and nuclear cost estimating procedures were investigated and utilized to determine capital as well as operating costs. An economically superior ship system, M2 (nuclear), was identified. The nuclear ship maintained the advantage when subjected to fuel oil, discount rate, and manning alterations. Finally, a few noneconomic elements were presented to remind the reader that EACF is not the sole decision parameter.

CHAPTER VI

OVERVIEW

The conclusion of this thesis will provide the reader with the opportunity to step back from the trees and evaluate the impact of the forest.

There are several basic assumptions that form the foundation of the thesis. These assumptions should be returned to the forefront of the reader's attention when evaluating the results.

1. Mission requirements

   --Break six feet of ice in the continuous mode

   --Achieve the following mission endurance profile:

   10 days at 50% power

   45 days at 80% power

   7 days at 100% power

   <u>15 days drifting</u>

   77 days = 48 full power days

2. Seek only an economic optimum, utilizing discounted cash flow techniques.

3. Satisfactory weight balance constitutes a feasible ship system (i.e. volume need not be considered due to increased efficiency of utilization as size increases).

4. POLAR STAR hull form constitutes the present optimum in icebreaking.
5. Design vessels to mission requirements.
6. Compare Nuclear-Steam Turbo-Electric propulsion with the optimum fossil fueled plant.

The results of the investigation conducted in this thesis indicate: The POLAR STAR hull form powered by a Nuclear-Steam Turbo-Electric propulsion plant, capable of delivering approximately 30,000 SHP, is the most economical ship system available to meet the mission requirements. The mission endurance profile is the driving parameter in the solution. With this in mind, the reader might question the value of the solution to a 10 year old problem. The response to the query is, that the U.S.C.G. is presently involved in the design of a polar icebreaker (approximately WIND Class length) with a 90 day mission profile. Nuclear power, providing independence from fuel oil storage capacity, allows the formulation of a ship system of significantly less displacement than the equivalent fossil ship system. In this particular case, the nuclear ship (M2) displays a 69 percent displacement advantage (12,862 versus 21,730 tons). This difference in overall size, reflected in capital and operating costs, plus the fuel cost advantage, more than compensate for the higher nuclear capital investment. Fuel sensitivity analysis, utilizing a 10 percent discount rate, indicates

that at least a 63 percent decrease in fuel oil price, to approximately $5 per barrel, is necessary to erase the nuclear cost advantage.

The investigations undertaken by the author have brought to light several areas worthy of further endeavor.

--Enlargement of Arctic environmental data base, particularly over the most viable trade routes.

--Enlargement of icebreaker, nuclear and conventional, data bases to validate, or otherwise, the suspicious linear nature of relationships contained in [11].

--Refinement of estimating procedures for nuclear application to icebreakers, removing dependence on merchant systems.

--Expansion of icebreaking modeling techniques to include volume, stability, and area balances.

--Investigation and, possibly, quantification of noneconomic parameters implicit in nuclear powered icebreakers.

--Education of operators with regard to the severity of impact of the ship-ice interface on powering, and, therefore, fuel and endurance capabilities.

Finally, the objectives of the thesis have been attained. Mission requirements were defined. Powering required to achieve the necessary icebreaking capability

was determined. Conventional (M1) and nuclear (M2) ship systems were formulated. An economic comparison, utilizing discounted cash flow techniques, selected the Nuclear-Steam Turbo-Electric powered icebreaker as the optimum ship system. Nuclear power provides a viable solution to the ever present endurance problems of a polar icebreaker. "In effect, use of nuclear power returns the operational limitation to the capabilities of the hull" [37].

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APPENDIX A

RAMMING-WHITE

White describes icebreaking in several stages or phases. He then formulates equations for each of these. The computer program simultaneously solves the equations. This Appendix presents the phase descriptions, an ice thickness calculation formula, and the White Bow.

"State 1" describes the icebreaker immediately prior to the initial contact with the ice. The "Crushing Phase" describes the local crushing of the ice to accommodate the ship's bow. The equations in this phase deal with three forces (i.e. normal to the bow plating, friction components acting both parallel, and perpendicular to the plating). As the bow rises, the velocity of the vessel decreases. "State 2" is the point when crushing has concluded and sliding has not begun. The "Sliding Phase" relates to the bow sliding up onto the ice with no further penetration into the ice sheet. "State 3" is the point at which the velocity on a point of the bow, relative to the ice, is zero. "State 4" is the condition when all velocities are zero. The downward force exerted by the vessel at this state determines the ramming capability of the vessel.

This computer program was run with U.S.C.G.C. WESTWIND input parameters. The results were compared with actual

structural strains measured during extensive testing in 1963, on board the WESTWIND. The program predicted a peak loading force occurring during the "Crushing Phase." This result agreed with the peak strain observed on the WESTWIND. The correlation with other predicted and measured variables was very good.

White [22] makes the next logical step, utilizing the vertical force component in the derivation of a formula for computing the maximum ice thickness that can be broken by ramming.

$$t_r = \left[\frac{F_r}{100}\right]^{1/2}$$

Parameters

F_r = $6.64\ (V)(WR*\Delta)^{0.845}$

V = impact velocity in feet/second

Δ = displacement in pounds

WR = White ratio

$= 0.000234(10.72 + B/T)(0.1833 + C_w)(1.652 - C_b)$
$(6.14 - (SAC)^2)(0.725 - f_k)(1.718 - B_A)$

B = beam in feet

T = draft in feet

C_w = waterplane coefficient

C_b = block coefficient

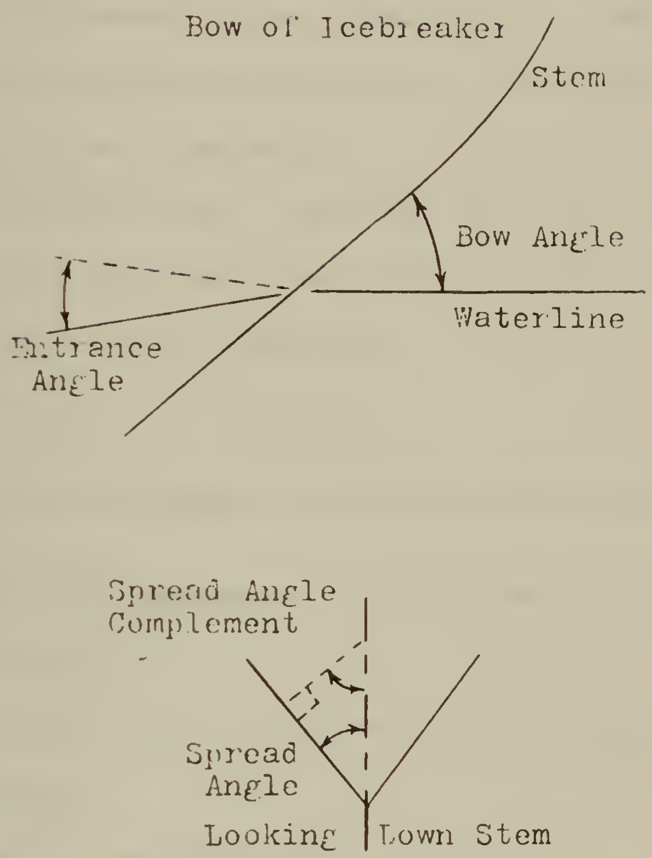
SAC = spread angle complement in radians (Figure A.1)

f_k = dynamic ice friction coefficient (assumed $f_k = 0.2$)

B_A = bow angle in radians (Figure A.1)

Figure A.1

Terminology of an Icebreaker Bow



The computer program allowed White to investigate the impact of many ship parameters. He concluded that the items contained in Table A.1, altered in the direction indicated, proved to have the greatest impact in increasing downward force generation. However, as the downward force is increased, so is the extraction thrust (the thrust necessary for the vessel to pull itself free of the ice). Reducing the static coefficient of friction and decreasing the spread angle complement, are beneficial to both problems.

Table A.1

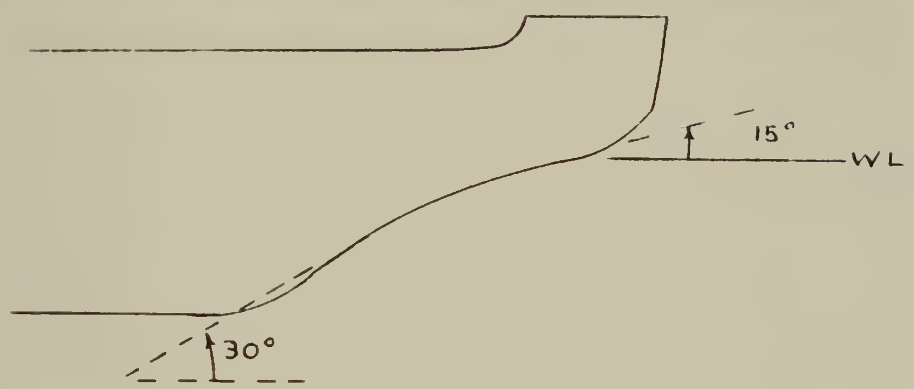
PARAMETERS FAVORABLE TO DOWNWARD FORCE GENERATION

1. Increase of displacement (approximately linear relationship).
2. Increase of impact velocity (approximately linear relationship).
3. Decrease of the bow angle.
4. Decrease of the spread angle complement (blunter bow).
5. Decrease of the coefficient of kinetic friction.
6. Decrease of the block coefficient.
7. Increase of the waterplane coefficient.
8. Increase of the beam-to-draft ratio.

Based on the above analysis the White Bow was derived (Figure A.2). The angle of initial entry should be small (15°-20°). The stem should be concave. The area in

Figure A.2

White Bow



contact with the ice in "State 4" should have a steeper slope (30°-35°); this aids extraction. The spread angle complement should be increased to approximately 0.6 radians (34°). The net effect of the changes will increase downward force, decrease peak loads on the vessel, and decrease the necessary extraction thrust.

APPENDIX B

SEMI-EMPIRICAL CONTINUOUS ICEBREAKING--VANCE

Vance develops the continuous mode icebreaking resistance equation, relating the pertinent physical and environmental parameters to the components of ice resistance, through dimensional analysis. This provides the basic form of the equation. The exponents and coefficients are found by subjecting five sets of model and full-scale data to a stepwise multiple regression analysis. Vance develops resistance equations for each vessel based on model data, validates each with corresponding full-scale data. The final equation is selected based on the least average error over the five vessels.

Vance begins by identifying the theoretical components of resistance:

1. Resistance due to icebreaking.
2. Resistance due to submergence of the ice floes.
3. Resistance due to turning of the ice sheet.
4. Resistance due to the loss of energy due to the changes in position of the icebreaker.
5. Resistance due to the impact of the hull with the ice.

6. Resistance due to pushing away broken ice.
7. Resistance due to the friction between the hull and ice, and between ice floes.
8. Resistance due to the viscosity of the water.

The important physical and environmental parameters are then described:

h - ice thickness. One of the most important parameters, entering into: the breaking of the ice sheet, the submergence and upturning of ice blocks, and the momentum exchange between ship and ice blocks.

σ_f - ice flexural or bending strength, entering only in the breaking component.

E - the modulus of elasticity of ice, entering into the bending of the sheet.

ρ_i - mass density of ice, entering the submergence and overturning, and possibly inertial components.

ρ_w - mass density of water, entering the submergence and inertial components.

f - coefficient of friction between hull and ice. This is a very complex parameter which may affect all components and must be given careful attention.

g - acceleration of gravity, entering into the submergence component.

ϕ - shape factor for the vessel, definitely affecting various resistance components. Due to the interrelation of all of the ship's angles, the shape must be assumed to affect all components.

B - beam of the ship, a very important parameter. This parameter enters into: the breaking, submergence, and pushing aside of broken ice block components.

T - draft of the vessel. No definitive studies have been made to date. However, it should have some effect on submergence.

L - length of the vessel. This is a relatively new parameter being considered in the resistance equation. The Arctic Marine Commerce Study (1973) proposed a resistance term including a length parameter. Johansson (1973) presents data dealing with Great Lakes ore carriers which indicate the significance of length. However, he does not include the length variable in his resistance equation. Vance is the first to analytically demonstrate its validity, and to include it in the velocity dependent term. Length enters into the inertial and friction components.

V - velocity of the ship, affecting the inertial term.

p - ice pressure. This parameter's effect is not definitely known, due to the lack of quantifiable data. It is felt that p plays a role in the friction component and should be included in the same term as the length parameter.

Resistance is, therefore, seen to be a function of the parameters described:

$$R = F(h, \sigma_f, E, f, \rho_i, \rho_w, g, \phi, B, T, V, L, p)$$

After defining the components and the significant parameters, Vance begins construction of the individual resistance terms.

The first component, resistance due to icebreaking, is modeled by a cantilever beam and also as an infinite wedge on an elastic foundation. The breaking resistance is shown to be a function of flexural strength, ice thickness, and the ship's beam:

$$R_B = f(\sigma_f, h, B) = f(\sigma_f^a\ h^b\ B^c)$$

Utilizing dimensional analysis techniques and present state of the art knowledge, the exponents were determined, to yield:

$$R_B = C_B \sigma_f h B$$

The second term in Vance's resistance equation is the result of combining submergence and upturning components. Separating the two interrelated components with a model test would prove extremely difficult and probably impossible. Vance models the phenomenon as a "beam of ice broken off by the icebreaker that is the beam of the ship wide and the length of unit travel long and as thick as the ice sheet." The submergence and upturn (R_S) is a function of the beam of the vessel, ice thickness, the draft of the vessel, and the difference in specific weight between water and ice:

$$R_S = f(B,h,T,\rho_\Delta g) = f(B^x,h^y,T^z,(\rho_\Delta g)^w)$$

$$R_S = C_S \rho_\Delta g B h^y T^z$$

where:

$\rho_\Delta g = (\rho_w - \rho_i)g$

y and z are investigated by regression analysis (found z = 0 and y = 2).

The third term in the resistance equation introduces the length and velocity parameters, and addresses the next four resistance components. Again, a physical model was employed by Vance. The side friction resistance was modeled as the normal force multiplied by the coefficient of friction, where the normal force is a function of side pressure, ice thickness, side angle, and length of the ship:

$R_f = fph \sin\phi\ L$

Velocity appears when the exchange of momentum between ice and vessel is considered:

$$R_V = C_V \frac{\rho_i}{2} BhV^2$$

Present model testing is not advanced enough to differentiate between R_f and R_V. Vance proposes that the four resistance components "are functions of the parameters included" in the R_f and R_V:

$$R_V = f(\rho_i, B, h, f, \phi, p, L, V)$$

Further investigation leads Vance to conclude that the velocity dependent resistance has the following form:

$$R_V = f(\rho_i, V, L, h, B) = f(\rho_i^k V^l L^m h^n B^p)$$

$$R_V = (C_V \rho_i V^2 L h^n B^p) p_f$$

where:

n and p are results of regression analysis (found n = .65 and p = .35).

p_f - side pressure factor (taken as 1).

The final term in the resistance equation is the open water resistance, which can be approximated as:

$$R_{OW} = \frac{1}{2} \rho_w S V^2 (C_f + \Delta C_f)$$

where:

$S = 16[\text{Displacement x Length}]^{1/2}$

$$C_f = \frac{0.075}{(\log_{10} R_n - 2)^2}$$

$\Delta C_f = 0.0004$

In summary, the resistance of a vessel, due to ice, in the continuous mode of icebreaking is the sum of three terms modified by a friction factor (C_f):

$R_{ice} = C_f(R_B + R_S + R_V)$

where:

$R_B = C_B \sigma_f Bh$--breaking

$R_S = C_S \rho_\Delta g B h^y T^z$--submergence

$R_V = (C_V \rho_i V^2 L h^n B^p)$--velocity, inertia

$C_f = 1$ (if coefficient of friction for model and full-scale are equal).

A stepwise multiple regression analysis was applied to full-scale and model data from five of the six vessels listed (in order of decreasing reliability):

1. U.S.C.G.C. MACKINAW
2. U.S.S.R. MOSKVA
3. Tug JELPPARI
4. Roll on/Roll off Ferry FINNCARRIER
5. U.S.C.G.C. STATEN ISLAND
6. U.S.S.R. ERMAK

The JELPPARI data was withheld from the regression analysis to be utilized as a test. The tug is not of large polar icebreaker stature, much smaller and of significantly different design. Model data was employed to obtain the coefficients, exponents, and terms of trial

resistance equations. Full-scale data was used to validate the derived equations. The output of this computer analysis was the "optimum regression equation (the one equation that has the least average error for all five ships)":

$$R_{(ice)} = C_S \rho_\Delta g B h^2 + C_B \sigma_f B h + C_V \rho_i V^2 L h^{.65} B^{.35}$$

The equation, obviously a compromise to fit the five ships, has a maximum loss in correlation from an equation that is optimum for a particular ship (MACKINAW) of 3.5 percent. The equation allowed prediction of full-scale resistance for large icebreakers to within 10 percent. The error increased to 15-20 percent for the smaller tug-like breaker JELPPARI. Figure B.1 illustrates correlation between full-scale data (FS), model scale regression (MSR), and full-scale regression (FSR), over a variety of ice thicknesses for the MACKINAW. Figure B.2 displays the predicted resistance curves utilizing Milano (1972), Edwards _et al_. (1972), and Vance (1974). Milano's approach is purely theoretical and does not account for the bow propeller effect.

Finally, Vance proposes an equation to be utilized if draft has been varied:

$$R_{(ice)} = C_s \rho_\Delta g B h^{2.5} T^{-.5} + C_B \sigma_f B h + C_V \rho_i V^2 L B^{.35} h^{.65}$$

The limited data available to date indicates the surprising decrease in resistance with increasing draft.

Figure B.1

Resistance Predictions - Vance

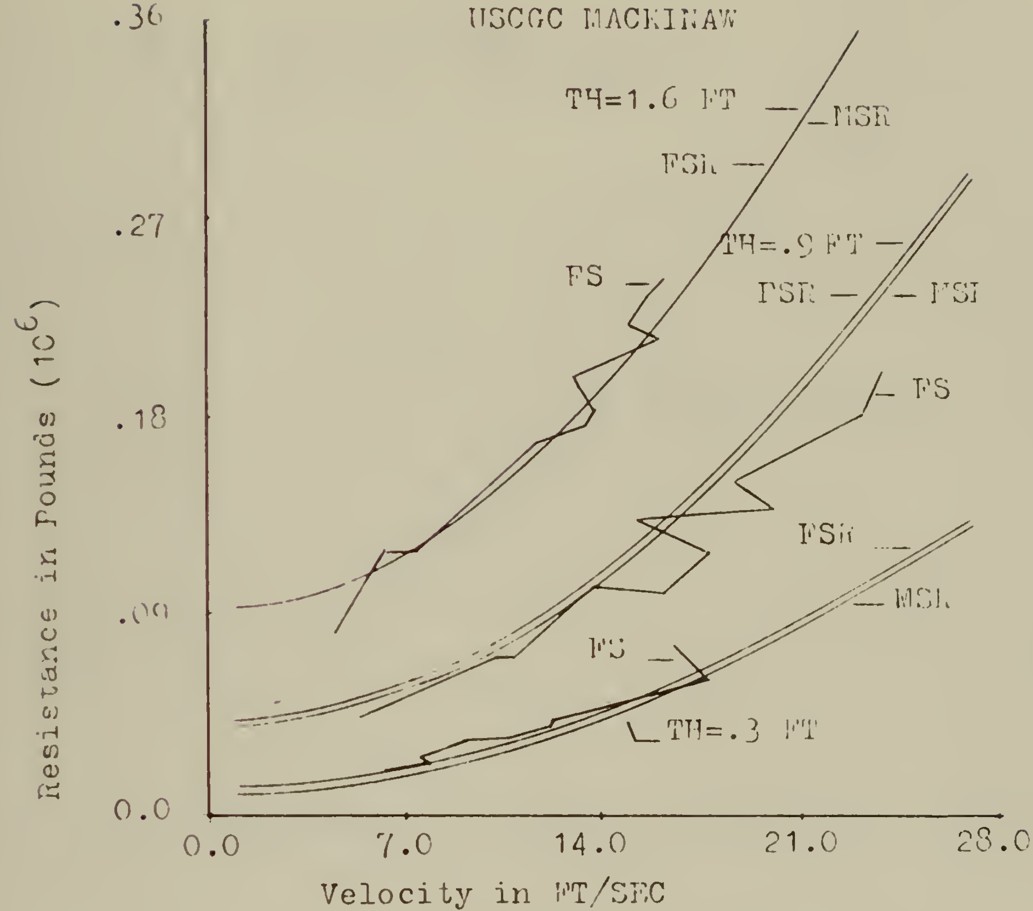
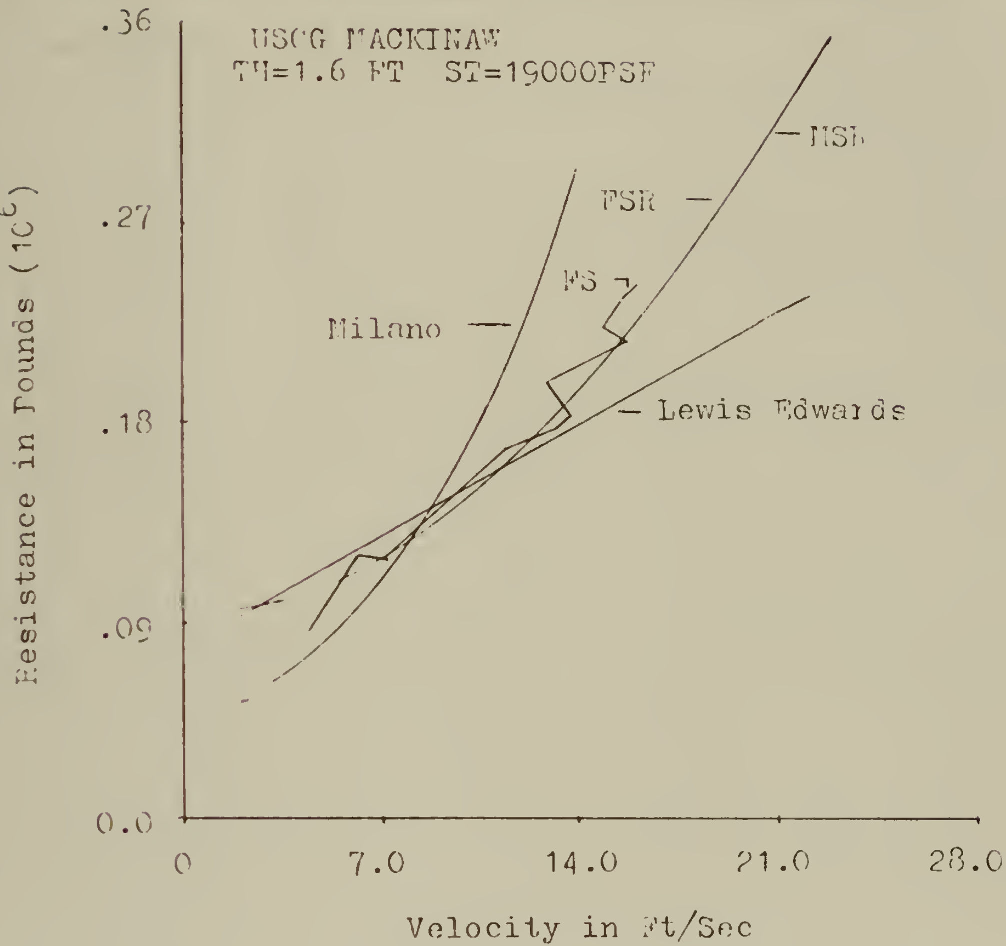


Figure B.2

Resistance Predictions - A Comparison



Vance has definitely provided a timely contribution to the art of icebreaker resistance prediction. His work certainly entails the most comprehensive use of existing data to date and is undoubtedly the best semi-empirical method available.

APPENDIX C

THEORETICAL CONTINUOUS ICEBREAKING--MILANO

Milano describes the continuous icebreaking process as the summation of five energies. The first of these energy components is due to ship motion through broken ice (E_1). This phase relates to the ship's resistance created as the ship comes in contact with broken pieces of ice. As they strike the hull, they acquire a certain velocity and move aside to allow the ship to pass. No actual breaking occurs, the resistance being primarily a resultant of inertial forces. The magnitude of this resistance is directly related to the ship size and the ice concentration. The model makes some simplifying assumptions:

1. The ship is symmetrical with respect to the center line.
2. The sides of the ship in way of the ice are vertical.
3. The ship's motion through the broken ice is at some constant velocity.
4. That the contact between ship and ice is inelastic.
5. The broken ice mass is of constant average density and continuous over the length of the hull to the maximum beam. Relief takes place aft of maximum beam, minimizing the resistance effect.

The program formulation becomes very critical in consideration of wall-sided vessels and vessels with maximum beam aft of midships. The program searches for the maximum beam forward of amidships and produces some strange unexplained resistance data if the maximum is aft. This problem appeared in the new U.S.C.G.C. 140 foot icebreaking tug model runs [17].

E_2 deals with the impact of the ship with the unbroken ice field. The basic assumptions of this phase are:

1. The surface hardness of the hull is much greater than the ice.
2. The energy loss of the ship upon impact is completely absorbed in local bending and crushing.
3. The ship is moving at a constant velocity prior to impact and strikes the ice in a "symmetrical wedging type manner."
4. Friction forces are secondary to impact forces and are, therefore, neglected.

E_3 relates to the ship's motion up onto the ice field. This results when the ice sheet does not fail under horizontal impact loading. The vessel must ride up, causing failure due to generation of a downward vertical force.

E_4 is concerned with the vessel's motion after the ice has failed, primarily falling. In the continuous mode, the change in draft is considered to be zero.

Therefore, the change in vessel position is related to pitching, yielding a loss of rotational kinetic energy.

E_5 discusses the resistance components due to ice slab rotation and submersion. As the ship moves forward, the broken ice cusps are driven downward and submerged. Buoyant and inertia forces are created which form a significant portion of the ship's resistance.

Milano seeks the total energy loss, the sum of E_1 through E_5. However, as the ice thickness varies, the significance of each energy loss term also varies. If the ice is very thick (but not enough to cause ramming), the energy loss is:

$$E_T = E_1 + E_{21} + E_3 + E_4 + E_5$$

where:

E_{21} = Impact of bow and cusps wedges, causing local crushing

For slightly thinner ice, "cusp wedge failure due to bending occurs before total available impact energy may be absorbed" [H] yielding a modified impact energy E_{22} and a reduced climbing energy term $E_3 = E_f + E_b$. Therefore, the total loss becomes:

$$E_T = E_1 + E_{22} + E_f + U_b + E_4 + E_5$$

where:

E_f = Total frictional energy loss at the bow and side cusps

U_b = Strain energy at bow and side cusps

For even thinner ice, impact causes failure, not only of cusps but also of bow edges, due to bending before total impact energy can be absorbed. E_2 is further reduced to E_{23}, and E_3 is reduced to only E_f. The total loss is now: $E_T = E_1 + E_{23} + E_f + E_4 + E_5$

Figure C.1 illustrates how the energy components add to give the full energy plot for a fixed ice thickness. Figure C.2 gives an indication of the importance of the different components due to increasing ice thickness. This can be seen in the similarity of curve shapes between Figures C.1 and C.2.

Utilization of this program requires definition of the constants in Table C.1, input parameters indicated in Table C.2, and provides the outputs listed in Table C.3. In addition to the outputs listed in Table C.3, the modification, performed by Vance, provides plots of resistance versus velocity and thrust versus velocity at constant SHP. If the curves are superimposed, the intersection will yield the speed made good through the ice.

The user should be aware of several foibles of the program. The sensitivity to the location of maximum beam has previously been mentioned. Investigation of the POLAR STAR data, contained in Table C.4, will indicate a sensitivity to the specific gravity of the ice (.82 to .92).

Figure C.1

Energy Loss Vs. Ship's Speed

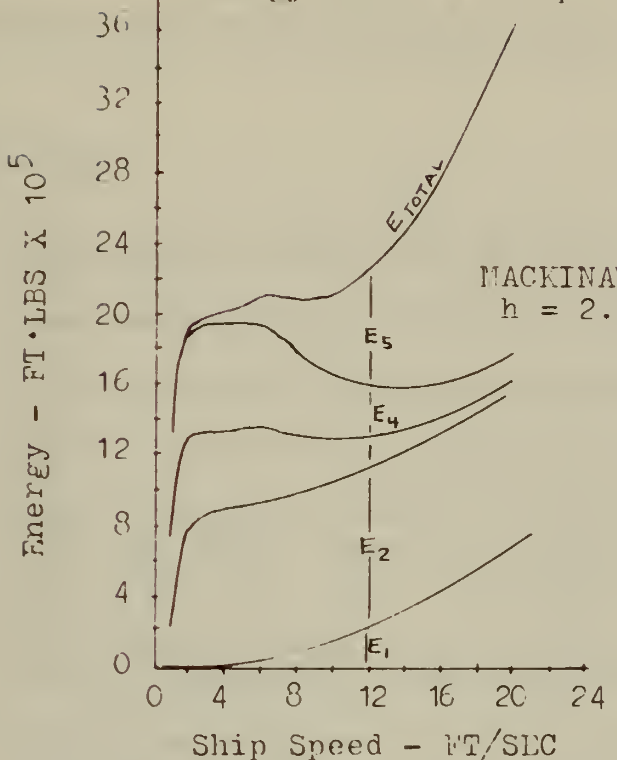


Figure C.2

Resistance Vs. Speed as a Function

of Ice Thickness

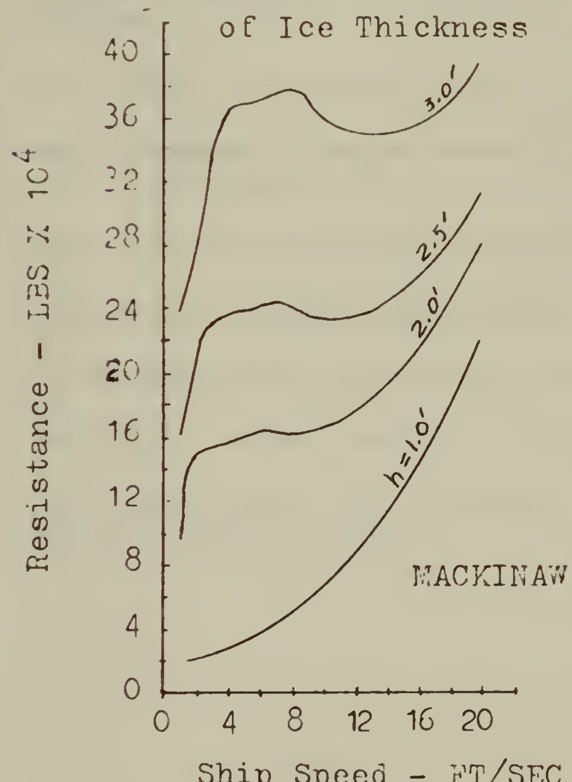


Table C.1

NOMENCLATURE FOR ICE RESISTANCE PROGRAM

Program Constants

RHOW	Mass density of water in LB SEC^2/FT^4 normally taken as 1.99
RHOI	Mass denisty of ice in LB SEC^2/FT^4 normally taken as 1.79
GAMW	Specific weight of water normally taken as 65.0 LB/FT^3
SPGR	Specific gravity of ice normally taken as .92
SIG	Tensile (flexural) strength of ice in LB/FT^3. In the area of 10,000 PSF for seawater ice and 20,000 PSF for fresh water ice. This value should be investigated by user.
SIGC	Compressive strength of ice in LB/FT^2. This value depends on ice temperature. 50,000 can be taken as an average value.
FRICT	Coefficient of dynamic friction. This value will vary and should be investigated by user. Range of values would be .1 to .5.
XNU	Kinematic viscosity of water normally taken as 1.97×10^{-5} FT^2/sec
GRAV	Acceleration of gravity normally taken as 32.2 FT/sec^2
DELCF	Hull roughness allowance normally taken as .004 (used for open water resistance)
PSIO	Bow wedge included angle normally taken as 1.18 radians
PSIC	Cusp wedge included angle normally taken as 1.85 radians
E	Young modulus for ice normally taken as 1×10^8 LB/FT^2
XK	Foundation (water) modulus normally taken as 64 LB/FT^2
ACCYF	An epsilon factor used in computing open water resistance normally taken as .001E-3

Table C.2

NOMENCLATURE FOR ICE RESISTANCE PROGRAM

INPUT

1. RLEN	Length between perpendiculars in feet
2. BX	Maximum beam in feet
3. DEL	Displacement in tons
4. SHP	Shaft horsepower per shaft in horsepower
5. PDIA	Propeller diameter in feet
6. X	Distance from bow to the section of maximum beam in feet
7. DELD	Distance from maximum beam to LCF in feet (aft is positive)
8. CBXL2	Distance from maximum beam to amidship in feet (aft is positive)
9. DRAFT	Draft in feet
10. CX	Midship section coefficient
11. CW	Waterplane coefficient
12. GML	Logitudinal metacentric height in feet at stated draft
13. ALPHA	Angle of inclination of the bow measured from horizontal radians
14. ALCG	Distance from LCG to LCF in feet (aft is positive)
15. NP	Number of propellers
16-18 CBI (I)	Station beam coefficient, $C_{BI} = B_I/B_X$ a 21 component array, from station 0 to 20
19-21 CXI (I)	Station area coefficient, $C_{XI} = A_{XI}/A_X$ a 21 component array, from station 0 to 20
22-23 GBETA (I)	Station transverse spread angle complement in radians an 11 or 15 component array station 0 to 10 or 14 taken from body plan depending on location of maximum beam
24-25 GDELT (I)	Station waterline inclination in radian an 11 or 15 component array station 0 to 10 or 14 taken from half-breadth plan depending on location of maximum beam
26-34 HICE	Ice thickness in feet to be investigated
35	0.0 - control data, indicates conclusion of data

Table C.3

NOMENCLATURE FOR ICE RESISTANCE PROGRAM

<u>OUTPUT</u>

HICE	Ice thickness in feet
U	Ship speed in feet per second
E1	Energy through ice filled channel in foot pounds
THRST	Total thrust available in pounds
TT1	Time to break
E3	Energy for climbing on ice in foot pounds
TT	Time in seconds
E4	Energy for fracturing ice in foot pounds
E5	Energy for submerging ice in foot pounds
E21	Energy for local crushing in foot pounds
ET	Total energy in foot pounds
FT	Total resistance in pounds
XFT	Non-dimensional resistance $(R/\rho gBH^2)$
XFROU	Thickness fraude number $(U/\sqrt{gh})$
XSIG	Non-dimensional strength $(\sigma/\rho gh)$

Table C.4

POLAR STAR COMPUTER OUTPUT

Fixed: h = 6 ft, V = 6 ft/sec, Draft = 28 ft

Run	σ_f(psf,10^3)	σ_c(psf,10^3)	R(lbs)	μ	S.G.	
14	10	100	325240.8	.2	.92	
16	12	100	400374.4			
17	15	100	534250.8			
15	20	100	813064.8			
34	15	52.5	570468.5			
32	12	50	428613.3			
36	14.3	50	537490.6			
21	15	50	570000			
37	16	50	627948.1			
43	15	50	564097.7 (FW)			(Draft = 30 ft)
44	15	50	566110.8 (SW)			(Draft = 30 ft)
38	18	50	762536.7			
30	20	50	860914.6			
35	15	45	581004.4			
27	30	50	1501141.7			
18	10	100	259264.1	.1	.92	
29	15	50	546263.4		.82	
39	15	50	461431.7		.92	
19	15	100	646534.9	.3	.92	
20	15	100	646534.9			
22	15	50	685999.9			
22A	15	50	685999.9			
28	15	50	450573.2	.2	.82 (spring ice)	

FW - fresh water

SW - salt water

S.G.-Specific gravity

Comparison of equivalent runs 21 and 28 shows a drop of 120,000 lbs (21%) in resistance for a shift in specific gravity from .92 to .82. However, Mother Nature rarely allows such a wide variance. Ice generally occurs in nature with a specific gravity ranging from .89 to .92. Therefore, the problem in the program provides no serious handicaps.

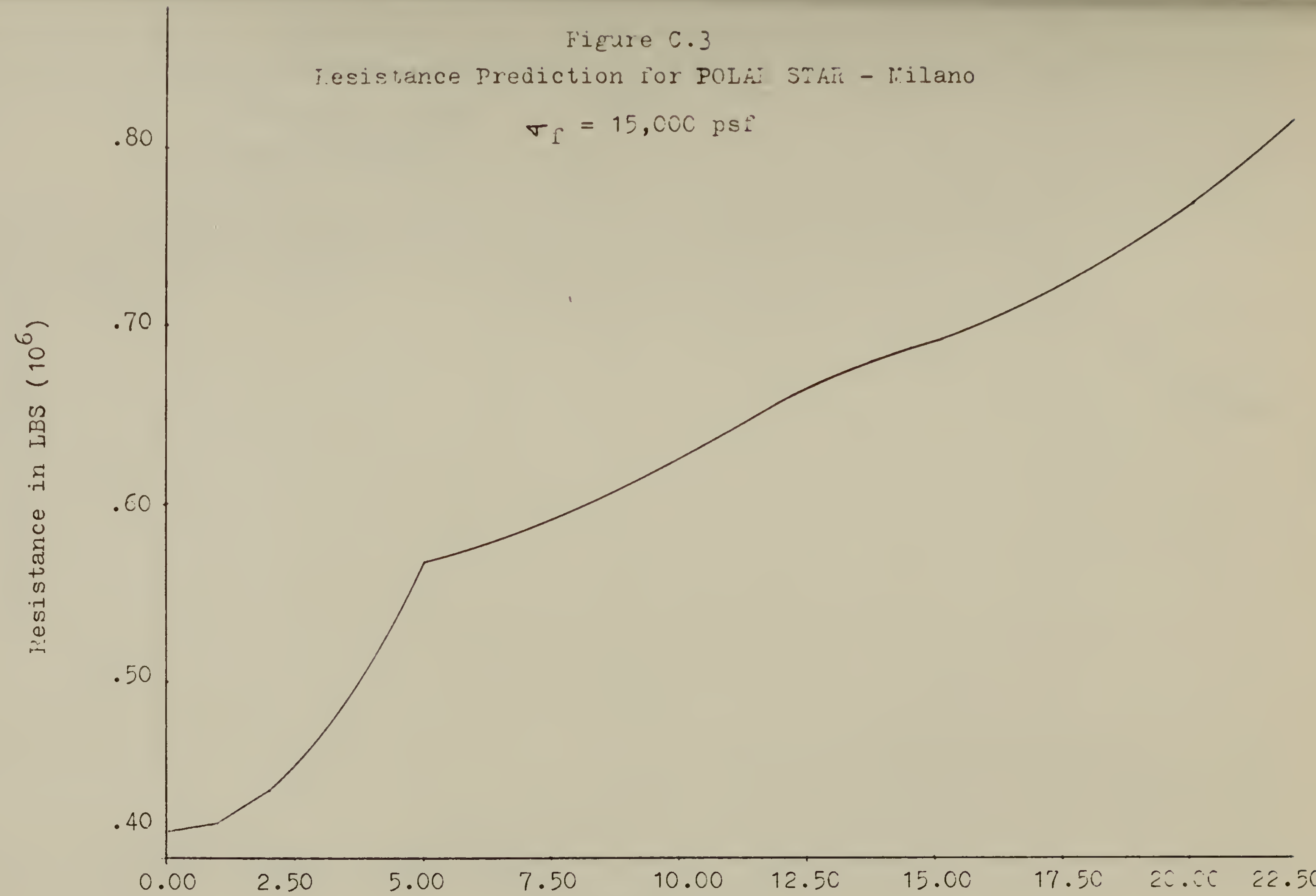
Milano's program also demonstrates extreme sensitivity to variations in ice flexural strength (σ_f), its resistance to breaking under a bending load. This is obvious from observation of the increase in resistance from runs 21 and 27 from 570,000 lbs to 1,501,000 lbs (163%). A comparison of Figures C.3 and C.4 shows the great distortion of the resistance curve. Fortunately, the user is again aided by Mother Nature. Flexural strength of sea ice ranges from 10,000 to 15,000 psf and 15,000 to 22,000 psf for fresh water ice.

One parameter, the coefficient of dynamic friction, has a significant impact on the resistance curves. Runs 39, 21, and 22 show resistance increasing by 24% (u = .1 to .2) and increasing 20% (u = .2 to .3). Vance [17] and Makinen et al. [20] feel that the coefficient of dynamic friction should increase resistance by 30% to 40%. This problem is not conveniently resolved by nature. The coefficient of dynamic friction is very difficult to measure accurately. Typically, this is accomplished by

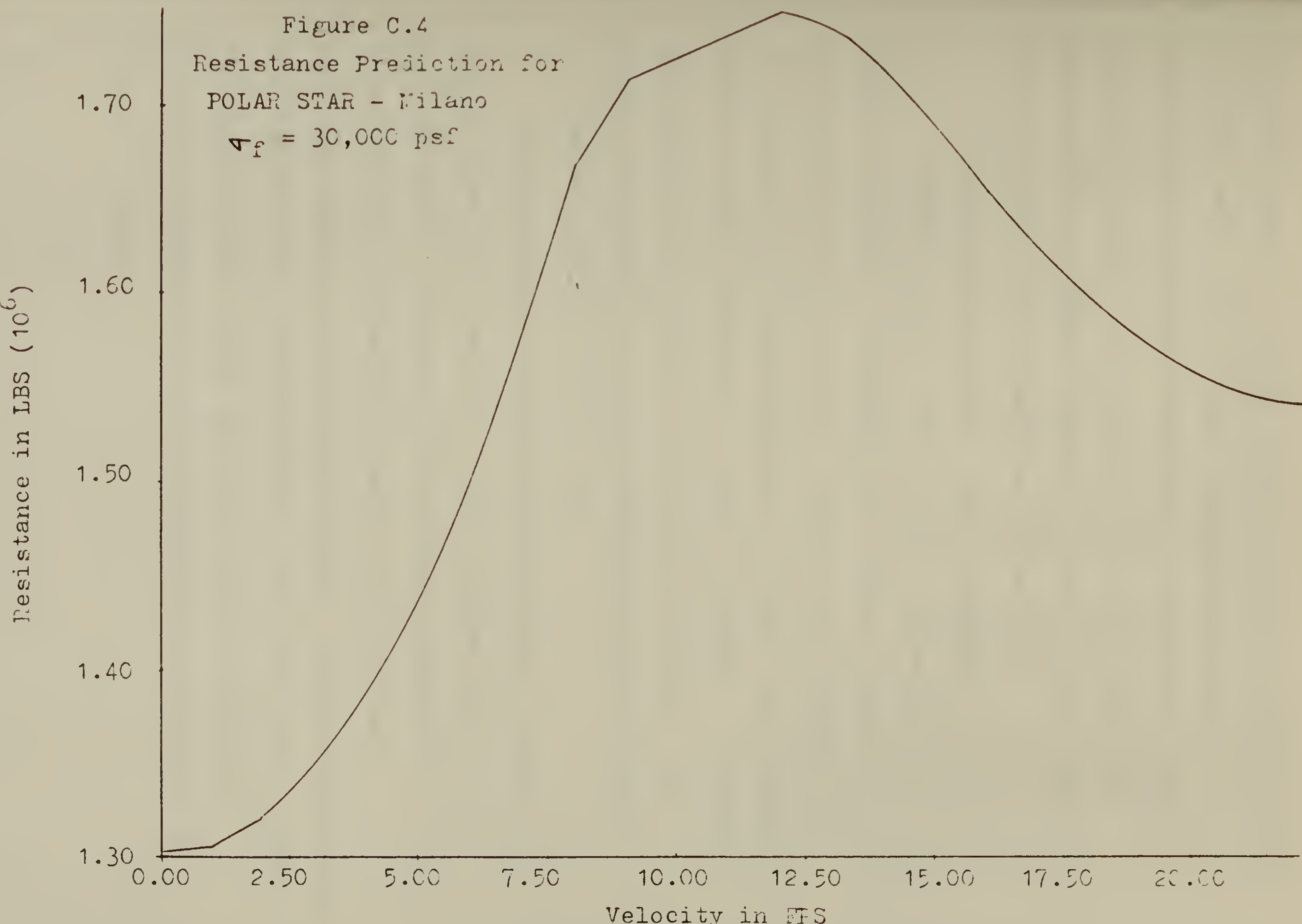
Figure C.3

Resistance Prediction for POLAR STAR - Milano

σ_f = 15,000 psf



Resistance vs. Velocity Ice=6 Feet



Resistance vs. Velocity Ice=6 Feet

dragging a piece of steel over the ice and measuring the required force. This rather rudimentary method of determining a parameter with such a significant impact is worthy of concern. At present, it is felt that the coefficient is probably between .1 and .2 [17]. Data outside of this range should be circumspect.

A parameter that Milano does not address is snow cover. Edwards et al. [15] indicate full-scale tests show "about two tons of resistance are added to total resistance by each inch of snow cover." Milano [9] proposes that "the motion of a ship through ice is such that lubrication of the ice/ship interface by water is common." Therefore, the effect of snow cover is difficult to judge. His program is probably slightly optimistic but definitely "a reasonable first estimate" [9].

After obtaining computer modeling data, the user will want to check the resistance curve against actual data. The user should be aware of the extreme sensitivity of the instrumentation necessary to sense the resistance variation below 6 to 8 feet per second. This is particularly true for the first knuckle between 2 to 4 feet per second. The ship designer should not find this too alarming, primarily because he is not interested in resistance at such low speeds.

APPENDIX D

MISCELLANEOUS DATA

Table D.1

POLAR STAR INPUT DATA BASE

1.	RLEN	352							
2.	BX	78							
3.	DEL	10863							
4.	SHP	10000							
5.	PDIA	16							
6.	X	140.8							
7.	DELD	33.3							
8.	CBXL2	35.2							
9.	DRAFT	28.0							
10.	CX	.852							
11.	CW	.740							
12.	GML	350.1							
13.	ALPHA	.26179							
14.	ALCG	1.35							
15.	NP	3							
16.	CBI (I)	0.0	.242	.426	.648	.789	.890	.955	
17.		.989	1.0	.999	.990	.974	.953	.925	
18.		.883	.823	.740	.622	.459	.249	.000	
19.	CXI (I)	0.0	.032	.150	.353	.556	.735	.856	
20.		.922	.973	.992	1.00	.968	.911	.824	
21.		.714	.636	.406	.267	.150	.032	.000	
22.	GBETA (I)	.524	.593	.716	.794	.922	1.114	1.257	
23.		1.295	1.309	1.326	1.335	1.344	1.335	1.274	
24.	GDELT (I)	.489	.471	.419	.353	.257	.175	.112	
25.		.056	0.000	0.000	0.000	0.000	0.000	0.000	
26.	1								
27.	2								
28.	3								
29.	4								
30.	5								
31.	6								
32.	7								
33.	8								
34.	9								
35.	0								

Table D.2

GENERAL PARTICULARS OF POST WAR ICEBREAKER CLASSES

Ship's Name	"WIND" CLASS	ABEGWEIT	EDWARD CORNWALLIS	THULE	D'IBERVILLE
Year Built	1944-47	1947	1949	1951	1953
Where Built	U.S.A.	Canada	Canada	Sweden	Canada
Nation	U.S.A.	Canada	Canada	Sweden	Canada
Length O.A.	269'0"	372'6"	259'0"	204'0"	310'6"
Breadth, Extr.	63.5'	61.08'	43.7'	52.8'	66.83'
Draft	25.75'	19.0'	18.04'	15.92'	30.42'
Displacement (Tons)	5,300	6,900	3,700	1,930	9,930
Speed/Knots	16.0(T)	16.0(T)	13.5	15.0	14.5(T)
Machinery	Diesel Electric	Diesel Electric	Steam Unaflow	Diesel Electric	Steam Unaflow
Total H.P.	10,000 S.H.P.	13,200 S.H.P.	3,500 I.H.P.	5,500 S.H.P.	10,800 I.H.P.

(T) indicates Trial Speed

Table D.2 (cont.)

Ship's Name	LABRADOR	GEN. SAN MARTIN	VOIMA	GLACIER	WILLIAM CARSON
Year Built	1953	1954	1954	1955	1955
Where Built	Canada	West Germany	Finland	U.S.A.	Canada
Nation	Canada	Argentina	Finland	U.S.A.	Canada
Length O.A.	269'0"	277'10"	274'0"	309'3"	351'0"
Breadth, Extr.	63.8'	62.4'	63.67'	74.0'	69.67'
Draft	30.1'	21.33'	22.12'	25.75'	19.4'
Displacement	6,490	4,830	4,415	8,300	7,720
Speed/Knots	16.0(T)	16.0	-	-	15.0(T)
Machinery	Diesel Electric	Diesel Electric	Diesel Electric	Diesel Electric	Diesel Electric
Total H.P.	10,000 S.H.P.	6,500	10,500 S.H.P.	21,000 S.H.P.	10,000 S.H.P.

(T) indicates Trial Speed

Table D.2 (cont.)

Ship's Name	MONTCALM	LENIN	KARU CLASS	MOSKVA	JOHN A. MACDONALD
Year Built	1957	1958	1958	1960	1960
Where Built	Canada	U.S.S.R.	Finland	Finland	Canada
Nation	Canada	U.S.S.R.	Finland	U.S.S.R.	Canada
Length O.A.	220'0"	440'0"	243'0"	400'6"	315'0"
Breadth, Extr.	48.3'	90.5'	57.0'	80.6'	70.25'
Draft	16.34'	30.25'	19.0'	35.0'	28.1'
Displacement	2,950	16,000	3,370	15,400	8,900
Speed/Knots	13.0	-	-	18.0	15.5
Machinery	Steam Unaflow	Nuclear Turbo-Electric	Diesel Electric	Diesel Electric	Diesel Electric
Total H.P.	4,000 S.H.P.	39,200 S.H.P.	7,500 S.H.P.	22,000 S.H.P.	16,500 S.H.P.

Table D.2 (cont.)

Ship's Name	TARMO CLASS	DANBJORN CLASS	FUJI	ST. LAURENT
Year Built	1963	1965	1965	1968
Where Built	Finland	Denmark	Japan	Canada
Nation	Finland	Denmark	Japan	Canada
Length O.A.	276'3"	247'0"	328'0"	366'6"
Breadth, Extr.	68.7'	55.9'	72.2'	80.0'
Draft	20.2'	19.8'	26.6'	29.5'
Displacement	5,850	3,685	8,566	13,300
Speed/Knots	-	17.0	15(T)	-
Machinery	Diesel Electric	Diesel Electric	Diesel Electric	Steam Turbo
Total H.P.	12,000 S.H.P.	10,500 S.H.P.	12,000 S.H.P.	24,000 S.H.P.

Table D.3

DIMENSIONS AND PRELIMINARY WEIGHT ESTIMATES

(LONG TONS) FOR DIESEL-ELECTRIC ICEBREAKER STUDIES

SHP	15,000	35,000	55,000	75,000
Length, B.P.	300'	400'	500'	600'
Breadth, D.W.L.	75'	90'	99'	103'
Draft	27.5'	30'	31.5'	33.1'
Depth	41.2'	46.5'	50'	51'
$L \times B \times D \times 10^{-5}$	9.3	16.8	24.7	31.6
Complement	350	450	550	650
Group 1, Long tons	2900	5025	7550	9600
2, " "	720	1675	2620	3460
3, " "	140	210	350	450
4, " "	55	85	120	150
5, " "	770	1350	1950	2400
6, " "	590	750	950	1100
7, " "	35	35	35	35
Lt. Ship Displacement	5210	9130	13575	17195
Comp., Stores, F.W.	650	810	1000	1170
Gas, Ammo., Avia. Eq.	100	150	230	300
Cargo	300	700	1100	1500
Lub. Oil	40	90	140	190
Fuel	2900	4400	7000	9300
Full Load Displacement L. Tons	9200	15280	23045	29655

Table D.4

INFLATION INDICATORS

	Labor Rates ($/hour)[*1]	Material[*2] Diesel (1141)	Group 1(1013)	Group 5(114)
1966	3.30	1.075	1.04	1.06
1967	3.34	1.15	1.075	1.1
1968	3.38	1.16	1.095	1.16
1969	3.68	1.19	1.105	1.19
1970	3.85	1.34	1.115	1.26
1971	4.00	1.53	1.3	1.36
1972	4.10	1.565	1.38	1.39
1973	4.45	1.61	1.42	1.44
1974	4.90	1.89	1.75	1.69
1975	5.35	2.26	2.04	1.94

[*1] From Bureau of Labor Statistics, "Employment + Earnings Index 3731", monthly publication. Wages are for average shipyard workers.

[*2] From Bureau of Labor Statistics, "Wholesale Price Index," monthly publication. Numbers are indices, inflation rates are found by looking at the annual change of the indices. Numbers in ( ) are item codes.

APPENDIX E

Caroussis provides a method for estimating the procurement and shipyard costs for a tanker. The "final shipyard bill" is composed of costs related to: hull structure (HS), outfit and hull engineering (O), and propulsion machinery (MN):

1. C_{HSM} - hull structure material

   = (Steel Weight)(Unit Steel Cost)

   = 5608 ton ($310/ton)

   = $1,738,480

   where: Unit Steel Cost = $310/ton

   Steel Weight (assume W_1) 5608 ton

2. C_{HSL} - hull structure labor

   = (Average Hourly Rate)(Hull Man-Hours)

   = ($6.15/hour)(333,221 hour)

   = $2,049,309

   where: Average Hourly Rate (AHR) = $6.15

   Hull Man-Hours = 70,600 $(\text{Steel Weight}/1000)^{.90}$

   = 333,221 HR

3. C_{OM} - outfit and hull engineering materials

   = (Outfit Weight)(Unit Outfit Material Cost)

   = 633 ton ($1500/ton)

   = $949,500

   where: Unit Outfit Material Cost = $1500/ton

   Outfit Weight (Assume W_6) = 633 ton

4. C_{OL} - outfit and hull engineering labor

$= AHR$ (Outfit Man-Hours)

$= (\$6.15/\text{hour})(204{,}000 \text{ hour})$

$= \$1{,}254{,}640$

5. C_{EA} = extra accommodation costs due to larger nuclear crew

$= \$150{,}000\ (PN - PF)^{.56}$

$= \$150{,}000\ (170 - 164)^{.56}$

$= \$409{,}125$

where: PN - number of personnel for nuclear ship

PF - number of personnel for fossil ship

6. C_{FKD} - first of kind development cost

$= \$8{,}670{,}000 + 15.9\ (SHP)$

$= \$9{,}147{,}000$

7. NSHIP - final shipyard bill for nuclear vessel

$= [(C_{HSM} + C_{OM})(1 + a) + (C_{HSL} + C_{OL})(1 + b)*$

$(1 + C) + C_{MN} + C_{EA}]*$

$[(1 + d)(1 + e)(1 + f)(1 + g)(1 + h)]$

$= [2{,}687{,}980(1.1) + 3{,}303{,}949(1.33)(1.7) +$

$21{,}328{,}246 + 409{,}125]*$

$(1.1)(1.6)(1.02)(1.03)(1.02)$

$= [2{,}956{,}778 + 7{,}470{,}229 + 21{,}328{,}246 + 409{,}125]$

(1.886)

$= \$60{,}663{,}211$ (1974 dollars)

$= \$69{,}843{,}314$ (1976 dollars)

where:	a - miscellaneous material cost	=	.10
	b - miscellaneous labor cost	=	.33
	c - overhead rate	=	.70
	d - profit	=	.10
	e - escalation costs	=	.06
	f - changes	=	.02
	g - owner's extras	=	.03
	h - engineering liaison	=	.02

APPENDIX F

The methods used to determine annual nuclear fuel costs for M2 power plant are described in this appendix. Caroussis [36] was the primary reference. Figure F.1 displays a simplified approximation for an "atom balance" typical of a thermal reactor utilizing low enriched uranium. Figure F.2 presents a detailed cash flow diagram followed by an equivalent cash flow for a single reactor core. Figure F.3 contains equivalent cash flows for the vessel's lifetime, and, finally, equivalent uniform annual cash flows.

To determine the cash flows, basic reactor core parameters, fuel cost parameters, and fuel cycle time parameters must be established. The basic reactor core relationships and assumptions are presented in Table F.1. Nuclear fuel cycle time parameter, necessary for discounted cash flow, are listed in Table F.2. Table F.3 lists the cost equations and assumptions. A complete calculation of annual fuel cost is performed in Table F.4.

Figure F.1

Simplified Fuel Reaction

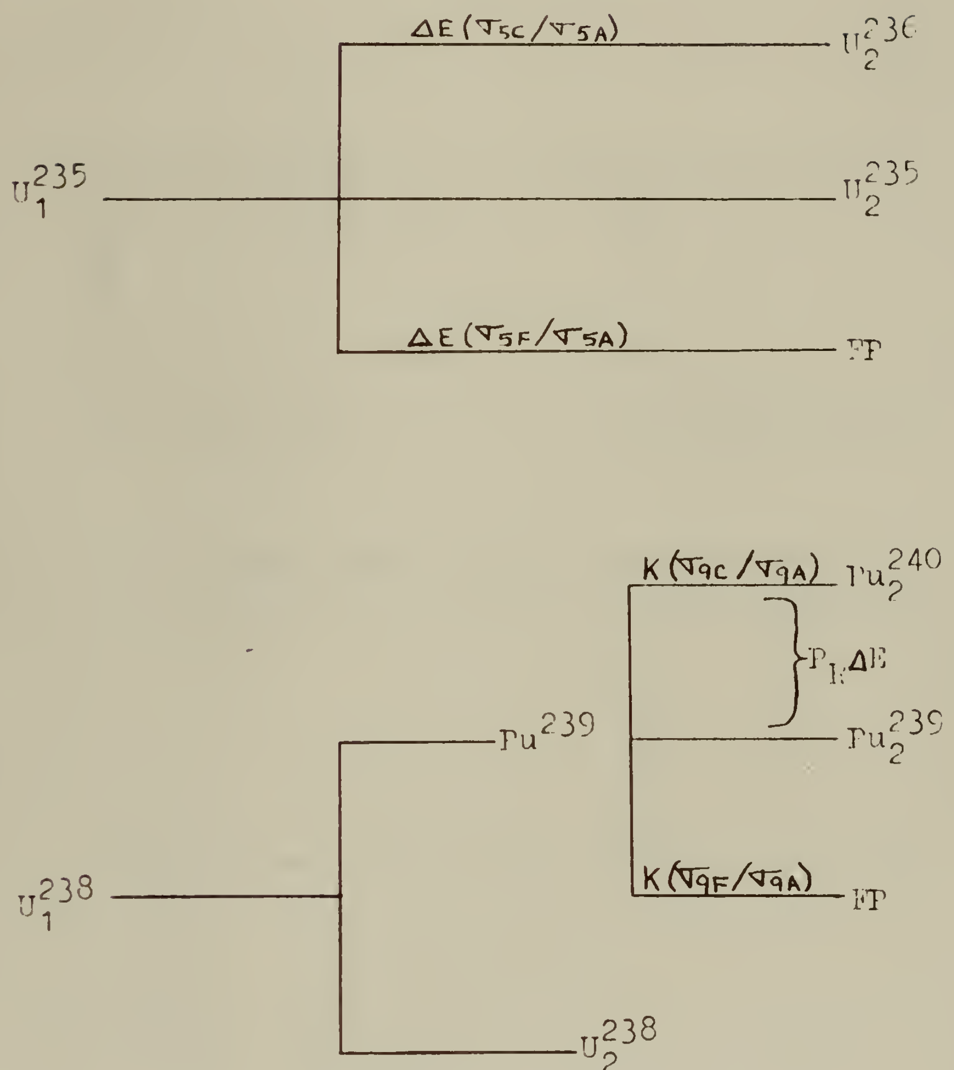
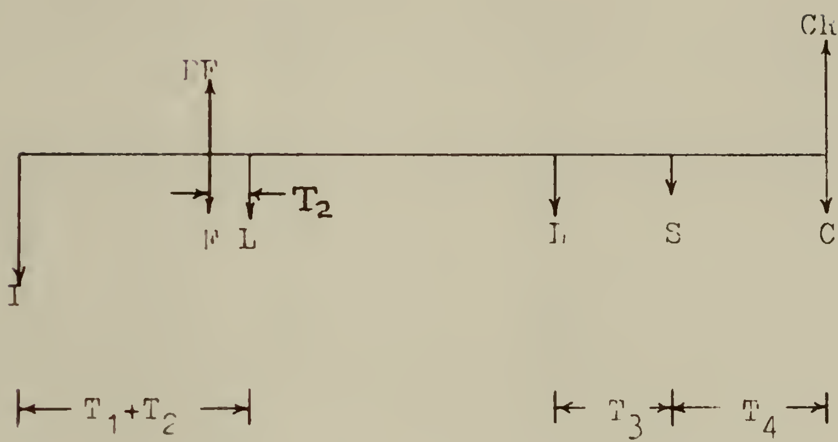
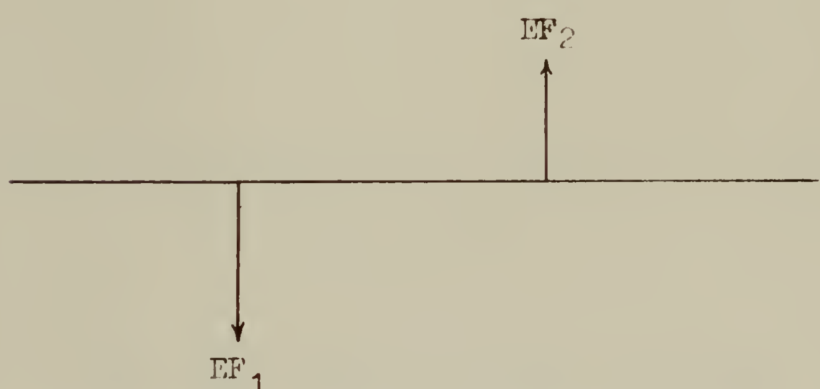


Figure F.2

Cash Flow per Reactor Core



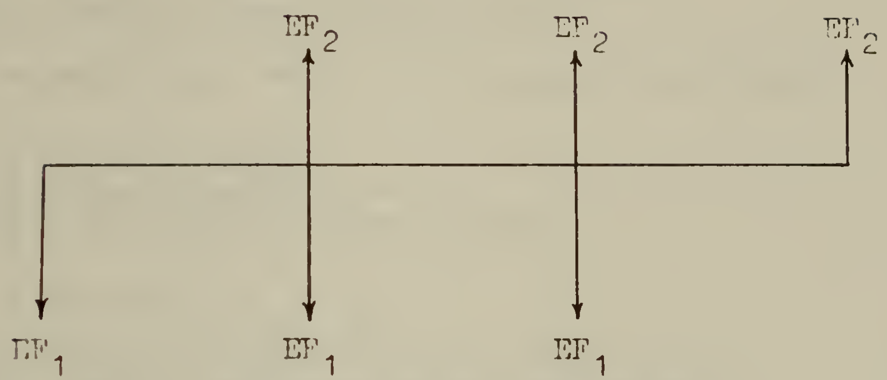
Actual Cash Flow per Reactor Core



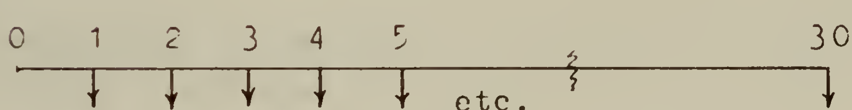
Equivalent Cash Flow (EF) per Reactor Core

Figure F.3

Equivalent Cash Flows



Equivalent Cash Flow Over Vessel Lifetime



Equivalent Uniform Annual Fuel Cost

Table F.1

BASIC CORE RELATIONSHIPS

$U_1 = 1000 * Mwt_m / S_p$ [Kg].

S_p = specific power - assume S_p = 18 KW/Kg. (1)
Mwt_m = reactor thermal megawatt rating

$BU = Mwt_N * 365 * (T_R/12) * PCF/(U_1/1000)$ [Megawatt-Days/Tonne] (2)

BU = reactor burnup
PCF = plant capacity factor
Mwt_N = normal reactor thermal output.

$W_{EP}/U_1 = \Delta E * [\sigma 5F/\sigma 5A) + C_R - P_R]$ (3)

W_{FP} = weight of fission products - [Kg.]
ΔE = change in enrichment during core residence
CR = plutonium conversion ratio - assume $C_R = 0.55$
P_R = plutonium buildup ratio - assume $P_R = 0.35$
$\sigma 5F$ = fission cross-section of U^{235}
$\sigma 5A$ = absorption cross-section of U^{235} } $(\sigma 5F/\sigma 5A) = 0.84$

$BU*(U_1/1000) = (1000*W_{FP}/235)*(6.02*10^{23}) *190* (4.45*10^{-23})/24$ (4)

Thermal energy absorbed per fission - assume = 190 MeV
Atoms per gram atomic weight = $6.02*10^{23}$
Megawatt-Hrs/MeV = $4.45*10^{-23}$

$E = 1.066 * BU * 10^{-6}$ (5)

From equations (3) and (4)

$Pu/U_1 = P_R * \Delta E - K * (\sigma 9C/\sigma 9A)$ (6)

Pu = fissile plutonium discharged
K = plutonium distribution coefficient
$\sigma 9C$ = capture cross-section of Pu^{239}
$\sigma 9A$ = absorption cross-section of Pu^{239} } $(\sigma 9C/\sigma 9A) = 0.28$

$K * (\sigma 9F/\sigma 9A = (C_R - P_R) * \Delta E$ (7)

$\sigma 9F$ = fission cross-section Pu^{239}
$\sigma 9A$ = absorption cross-section of Pu^{239} } $(\sigma 9F/\sigma 9A) = 0.72$

$Pu = 2.88 * BU * U_1 * 10^{-7}$ (8)

From equations (5), (6) and (7)

Table F.1 (cont.)

$$U_2 = U_1\ (1 - E * (\sigma 9F/\sigma 9A) - C_R * \Delta E) \qquad (9)$$

$$U_2 = (1 - 1.482 * BU * 10^{-6})\ U_1$$

From equation (5)

$$E_2 = E_1 - \Delta E \qquad (10)$$

$$E_2 = E_1 - 1.066 * BU*10^{-6}$$

From equation (5)

Table F.2

NUCLEAR FUEL CYCLE TIME PARAMETERS

T_1 = Fuel Fabrication Period

Shipping: AEC to fabrication site	0.75 months
UF_6 to UO_2 conversion process delay	0.50
UF_6 to UO_2 conversion process	1.00
Fabrication process delay	1.00
Fabrication process	1.00
Shipping: fabrication site to reactor site	0.75
Total	5.00 months

T_2 = Fuel Storage Period

Storage at refueling site	1.00 months

T_R = Fuel Residence Time

$T_R = 12 * (LE/NC) - T_F$
LE = vessel economic life
NC = number of reactor cores in vessel economic life

T_F = Refueling Period

Refueling time	0.50 months

T_3 = Fuel Cooling Period

Cooling Period	4.00 months

T_4 = Fuel Reprocessing Period

Shipping: discharge site to reprocessing site	0.75 months
Separation process delay	0.75
Separation process	0.25
$U(NO_3)_4$ to UF_6 conversion process delay	0.25
$U(NO_3)_4$ to UF_6 conversion process	0.25
Shipping: reprocessing site to AEC	0.75
Total	3.00 months

Table F.3

NUCLEAR FUEL COST RELATIONSHIPS

I = Cost of Enriched Fuel Material [$]

$V_1 = (p_1 + 0.5) * (1 + 0.12) * U_1$

p_1 = unit price of enriched uranium [$/Kg]

$p_1 = [- 89.714 + 126.324 * E_1 + 0.8045 * E_1{}^2]$ (least squares fit)

E_1 = initial fuel enrichment - U^{235} per U - kg/Kg.

U_1 = initial uranium inventory [Kg.]

Assumes $0.5/Kg. U shipping to fabrication sight
Assumes 12% purchase margin due to:
Assumes 10% fabrication scrap recycle
Assumes 1% loss during fabrication
Assumes 1% loss during conversion to UF_6 to UO_2

RF = Credit for Recycle Scrap Fuel Material [$]

$RF = (p_1 - 0.5) * 0.10 * U_1$

Assumes $0.5/Kg. U Shipping cost to AEC
Assumes 10% fabrication scrap recycle
Assumes no charge for converting scrap UO_2 to UF_6

F = Cost of Converstion, Fabrication and Shipping of New Fuel Elements [$]

Fuel = $80 * U_1

Assumes $80 Kg. U converting UF_6 to UO_2 + fabrication + shipping to refueling site

L = Cost of Core Loading or Removal = $250,000 [11]

Table F.3 (cont)

S = Cost of Shipping Spent Fuel Elements [$]

$S = \$4.0 * U_2$

U_2 = discharge uranium inventory - [Kg]
Assumes $/Kg. U shipping cost to reprocessing site

C = Cost of Separation, Conversion and Shipping of Spent Fuel Material [$]

$^*C = (31{,}300/b + 5.60 + 0.5) * U_2$

b = daily separation batch size of:

E_1	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10
b	1000	880	740	650	590	540	500	465

$(b = 231.76 + 177.17 * E_1 - 15.01 * E_1^2)$ - least squares fit

Assumes $5.60/Kg. U cost of converting $U(NO_3)_4$ to UF_6
Assumes $0.5/Kg. U shipping cost to AEC

CR = Credit for Depleted Fuel Material and Fissile Plutonium Produced

$CR = p_2 * 0.987 * U_2 + \$10{,}000 * 0.99 * Pu$

P_2 = unit price of depleted uranium - [$/Kg.]
(AEC cost data - Reference 36)

Assumes 1% uranium and plutonium loss during separation
Assumes 0.3% uranium loss during conversion from $U(NO_3)_4$ to UF_6
Assumes $10.0/g credit for fissile plutonium in $Pu(NO_3)_4$ form

*For $U_2/b \le 24$, $C = (23{,}500/b + 5.6 + 0.5) * U_2 + \$188{,}000$

Table F.4

SAMPLE ANNUAL NUCLEAR FUEL CALCULATION

Let: r = discount rate = 10%

LE = 30 years

NC = 3

SHP = 30,000

E_1 = 4.7% w/o (weight percent)

Reactor Calculations

$Mwt_m = 3.565\ (SHP/100)^{.965} = 94.95$ Mwt

$Mwt_N = 1.096337\ Mwt_m = 86.61$ Mwt

PCF = (200 days/365days) (48 full power days/77 days) = .34

$U_1 = 1000\ (94.95/18) = 5275.0$ kg

$BU = Mwt_N\ (365)\ (T_R/12)\ PCF/(U_1/1000) = 19097.4$ Mwd/Tonne

$E_1 = 1.1 + [.144 + 2.72/Mwt_m + 143.57/(Mwt_m)^2]\ BU/1000$

$= 4.70$ [36]

$\Delta E = BU(1.066E\text{-}06) = 2.04$

$E_2 = 4.70 - 2.036 = 2.66$

$PU = BU(U_1)\ (2.88E - 07) = 29.01$ Kg

$U_2 = U_1(1 - BU(1.482E - 06)) = 5125.7$ Kg

Fuel Cycle Cash Flows

$P_1 = .8045\ E_1^{\ 2} + 126.324\ E_1 - 89.714 = \$521.76$

$I = (P_1 + .5)\ (1.12)\ U_1 = \$3,085,520$

$RF = (P_1 - .5)\ (.10)\ U_1 = \$274,965$

$F = (80.0)\ U_1 = \$422,000$

$L = \$250,000$

$S = 4.0(U_2) = \$20,503$

$b = 231.76 + 177.17\ E_1 = 15.01\ E_1^{\ 2} = 733$

$U_2/b = 5125.7/733 = 7$ which is less than 24

$C = (23,500/b + 5.6 + 0.5)\ U_2 + 188,000 = \$383,600$

$CR = P_2(.987)\ U_2 + 10,000\ (.99)\ Pu$

$P_2 = (.8045\ (E_2)^2 + 126.324\ (E_2) - 89.71)$ [36]

$CR = \$1,562,000$

<u>Annual Equivalent Cycle Costs</u>

All costs are in millions

1. $R = r + 1 = 1.10$ (10% discount rate)

$$EF_1 = (I)R\ \exp((T_1 + T_2)/12) + (F - RF)\ R\ \exp(T_2/12) + L$$

$$= 3.085\ (1.1)^{1/2} + (.422 - .275)\ (1.1)^{1/12} + .250$$

$$= 4.968$$

$$EF_2 = (CR - C)\ [R\ \exp-(T_3 + T_4)/12] - (S)\ R\ \exp(-T_3/12) - L$$

$$= (1.562 - .384)\ (1.1)^{-7/12} - .021(1.1)^{-1/3} - .250$$

$$= 1.114 - .020 - .250$$

$$= .844$$

$$NPV = EF_1 + (EF_1 - EF_2)\ [\ R\ \exp(-30(1/3))] +$$

$$(EF_1 - EF_2)\ [R\ \exp(-30(2/3))] - EF_2\ (R\ \exp(-LE))$$

$$= 4.968 + (4.124)\ (1.1)^{-10} + (4.124)\ (1.1)^{-20} -.844(1.1)^{-30}$$

$$= 4.968 + 1.590 + .613 - .048$$

$$= 7.123$$

Annual Fuel Equivalent Cost = Total NPV $[r/(1 - R\ \exp\ (-LE))]$

$$= 7.123\ [(.1)/(1 -(1.1)^{-30})]$$

Table F.4 (cont.)

$= \$756{,}000/\text{year}$

2. $r = .20$ (20% discount rate)

$R = 1.20$

$EF_1 = 3.085\ (1.2)^{1/2} + (.147)\ (1.2)^{1/12} + .250 = 3.779$

$EF_2 = 1.178\ (1.2)^{-7/12} - .021\ (1.2)^{-1/3} - .25 = .7894$

$$NPV = 3.779 + 2.9896\ (1.2)^{-10} + 2.9896\ (1.2)^{-20} - .7894\ (1.2)^{-30}$$

$$= 4.336$$

$$\text{Annual Fuel Cost} = 4.339 \frac{.2}{(1-(1.2)^{-30})}$$

$$= \$871{,}471/\text{year}$$

Thesis
G8637 Gulick 165476
Nuclear vs. fossil
propulsion for ice-
breakers.